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SUMMARY REPORT

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on

SPACE-VEHICLE STABILIZED-PLATFORM
GIMBAL-SYSTEM WEIGHT-REDUCTION STUDY

PHASE V. GIMBAL-SEAL CARTRIDGE DESIGN

to

GEORGE C. MARSHALL SPACE FLIGHT CENTER

March 31, 1964

NAS 8-5101

by

J. W. Adam, T. J. Atterbury, and T. M. Trainer

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BATTELLE MEMORIAL INSTITUTE
505 King Avenue
Columbus, Ohio

June 4, 1964

Director
George C. Marshall Space Flight Center
Huntsville, Alabama

Attention M-P & C-CA

Dear Sir:

NAS 8-5101

We are enclosing nine copies and one reproducible master of the report "Space-Vehicle Stabilized-Platform Gimbal-System Weight-Reduction Study", Phase V, "Gimbal-Seal Cartridge Design", covering the work performed on the fifth phase of this program.

This report summarizes the experimental work performed in the process of determining design information for a gimbal air seal.

If you have any questions regarding these results, or would like to discuss any part of this program, please call.

Sincerely,

T. J. Atterbury, Director
Applied Solid Mechanics

TJA:so
Enc. (10)

ABSTRACT

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Phase V of the program was undertaken to develop a seal cartridge for the transmission of low-pressure gas between the gimbals of a stabilized platform, with primary objectives of minimum torque and minimum leakage.

This report describes the investigation of candidate seal materials and experiments on single- and double-seal characteristics. The work showed that the torque and leakage of present gimbal configurations can be significantly reduced if a bellows-actuated, Teflon face seal is used. The mating-seal materials should be virgin Teflon finished by the cold-press procedure, and a concentrically ground metal surface. The Teflon seal should be approximately 1/8 inch wide and of overbalanced design.

This research program was initiated in July, 1962, under contract with NASA. This report covers the work performed on Phase V during the period August 1, 1963, to March 31, 1964.

- *Author*

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SPACE-VEHICLE STABILIZED-PLATFORM GIMBAL-SYSTEM WEIGHT-REDUCTION STUDY

PHASE V. GIMBAL-SEAL CARTRIDGE DESIGN

INTRODUCTION

The importance of achieving minimum weight in the final stages of ballistic missiles or in the payload of space vehicles is well known. One method of achieving minimum weight is by the exacting design of each component on the basis of stresses and deflections. Another method, and one which is sometimes overlooked, is by a critical re-examination of the functional requirements of each of the components. It was the application of the second of these two methods which led to Phase V of the over-all weight-reduction study for the Saturn stabilized platform.

In the Saturn vehicle the gyroscopes of the stabilized platform gyros are supported on air bearings. This air must be supplied from an external source, which means that each of the pivots must contain a dynamic air seal. The size, and hence the weight, of the torque generators located at each pivot is dependent on the torque required to overcome the friction of the dynamic air seal and of the roller bearings. The major portion of this torque results from the air seal. Hence, if the torque and leakage of the seal could be reduced, a major reduction would be possible in the size and hence the weight, not only of the torquers, but of the external air supply.

The original objective of Phase V of the over-all weight-reduction study was to develop a seal cartridge for the transmission of low-pressure gas between the gimbals of the stabilized platform with minimum torque at the gimbals and with minimum weight. This objective was modified early in the project when information on seal materials was found to be inadequate. It was mutually agreed with the technical monitor that more emphasis should be placed on obtaining basic design data and that the work on the development of a cartridge configuration should be reduced.

The first step in the research effort was to review the functional requirements and boundary conditions for the seal cartridge. This included a materials investigation and a review of some of the characteristics of commercial seals. These studies resulted in the above-mentioned change in project emphasis. The next step was a single-seal interface experimental study to determine the effect on torque and leakage of seal parameters such as width, mating-ring surface characteristics, seal surface finish, etc. Once these effects were established, a double-seal experimental study was performed to determine the interaction of opposed face seals loaded by means of a common bellows.

The following section of this report presents the conclusions and recommendations resulting from the research effort. The next section describes the studies of possible seal materials. The single-seal interface experimental study and the double-seal experimental study are then discussed. In the final section of the report, seal cartridge characteristics are presented.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The mating-seal materials should be virgin Teflon finished by the cold-press procedure described in the report, and a concentrically ground metal surface with a surface finish of approximately 5 to 12 microinches rms. The width of the Teflon seal should be approximately 1/8 in. A sealing load of 4.7 lb is optimum for a mean seal diameter of 1.373 in. This load includes the load provided by the bellows and the load from the air pressure caused by the overbalanced design.

The torque and leakage values for the experimental double-seal units at 15 psi ranged from 8.4 to 13.8 oz-in. and from 15 to 65 cc per min at seal loads of 3.9 to 5.5 lb. These torque and leakage figures, which are representative maximum and minimum values, are a significant improvement over the "guide" values of 34 oz-in. and 500 cc per min which were provided at the beginning of this program as being typical of present seals.

Recommendations

It is recommended that a Teflon face seal be incorporated into a cartridge design. The cartridge concept is recommended because of the high reliability of a pretested seal, the necessity for run-in, and the elimination of damage to the sealing surface due to handling.

MATERIALS INVESTIGATION

The seal cartridge had to satisfy several functional requirements that severely restricted the candidate materials for the face seal. One of the most important requirements was that the mating materials have a very low coefficient of friction. Because of the downstream air bearing and the close tolerances involved, it was imperative that no contamination be introduced into the air flow. Therefore, the material had to exhibit a low coefficient of friction with dry air as the only lubricant. An upper temperature limit of 60 C did not appear difficult. However, the difference between the static and kinetic coefficients of friction had to be taken into account because of possible undesirable stick-slip characteristics. Considering these requirements, Teflon was a logical choice. The materials investigation, therefore, had three major aims: (1) to compare Teflon with other materials, thus insuring that Teflon was indeed the proper choice, (2) to compare various Teflon compounds to determine which of the fluorocarbons should be used, and (3) to determine the characteristics of the Teflon compound chosen.

Two of the most common materials used as face seal materials in commercial applications are carbon graphite and the industrial ceramics such as aluminum oxide. The major drawback of these two materials for the gimbal-seal application was the high coefficient of friction when used in the nonlubricated condition. The facet that these

materials can be machined to close tolerances with good surface finishes, are inert to most industrial chemicals, and have good heat-transfer characteristics, is the basis for their choice for commercial face seals. For the gimbal-seal cartridge, however, the extremely low rotational speed negated the requirement for good heat-transfer characteristics. Hence, organic materials could be considered for the face seal.

Table 1 gives some typical coefficient-of-friction values for different plastics when mated with carbon steel.^{(1)*} As can be seen, all of these values are higher than that obtained with Teflon with the exception of Rulon, which is a glass-filled Teflon compound. The large spread between the maximum and minimum values shown in the table might be explained by the extreme range of applied loads and sliding speeds which were used in the collection of the data. Other investigators⁽²⁾ indicate that the low coefficient of friction for Teflon and Teflon compounds holds only at very low sliding speeds, and then only if there has been no high-speed sliding. Once the plastic has undergone a high-speed sliding, the coefficient of friction is irreversibly increased to two or three times its previous value.

The use of reinforcing fibers generally results in a higher coefficient of friction for the filled compound, with the increase in coefficient of friction proportional to the filler content.⁽³⁾ Coefficients of friction for typical filled compounds are given in Table 2.^(4,5) The increase with filler content can be seen by comparing the coefficients of friction for unfilled and 15 and 25 per cent glass fiber. Normally the major reason for choosing a Teflon compound which has been filled is to decrease the amount of creep occurring under load conditions. With the low loads necessary to achieve a minimum torque in the cartridge seal, this tendency to cold flow was of secondary importance.

Use of the nonfibrous fillers such as graphite and bronze was out of the question because of the possibility of particles breaking off and contaminating the air bearing in the gimbal system. Teflon does not flake or powder under stress and retains its mechanical properties practically unchanged to temperatures around 300 C.⁽⁶⁾ In addition, the use of some of the familiar dry lubricants such as molybdenum disulfide and graphite as fillers cannot be expected to reduce the coefficient of friction below that of pure Teflon because normally they have coefficients of friction greater than that of Teflon.⁽¹⁾

Fluorocarbons are divided chemically into two major categories: polytetrafluoroethylenes and polytrifluoroethylenes. Of these, CTFE is the major commercial product from the polytrifluoroethylenes and exhibits coefficients of friction on the order of 0.33 to 0.36 and 0.27 to 0.34 for static and kinetic conditions, respectively.⁽⁷⁾ CTFE is thus eliminated from consideration. There are two major types of polytetrafluoroethylenes: TFE (tetrafluoroethylene) and FEP (fluorinated ethylene propylene). Figures 1 and 2 show the static and kinetic coefficients of friction for TFE and FEP.⁽⁸⁾ As can be seen, TFE has the lower coefficient of friction under comparable loading and speed conditions. These figures are typical of the type of data which are available for Teflon. That is, most of the data do not extend to the low loads and low speeds which are of interest for the gimbal seal. In general, most of the data available indicate that the coefficient of friction for Teflon increases below loadings of 30 to 50 psi, decreases rapidly at rubbing speeds below 25 to 50 fpm, and is relatively independent of temperature in a range of 80 to 620 F.⁽⁹⁾

The properties of TFE given in Table 3 are general and apply to all of the TFE resins shown in Table 4.^(7,8) Unless otherwise specified, Teflon 1 is normally the compound supplied in sheet and rod. This is the compound which was used in the experimental program described later in this report.

*Numbers in parentheses refer to references listed at the end of the report.

TABLE 1. TYPICAL COEFFICIENTS OF FRICTION
FOR VARIOUS PLASTICS⁽¹⁾

Plastic	Coefficient of Friction ^(a)	
	Max	Min
Teflon	0.21	0.09
Rulon	0.19	0.12
Cellulose acetate	0.53	0.18
Ethyl cellulose	0.57	0.22
Kel-F	0.25	0.17
Methacrylate	0.47	0.16
Nylatron G	0.33	0.22
Nylon	0.33	0.15
Polyethylene	0.40	0.17
Polystyrene	0.45	0.12

(a) No lubrication, 1-5 lb applied load, crossed cylinder apparatus, 8-365 ft/min sliding speed.

TABLE 2. COEFFICIENTS OF FRICTION
FOR TYPICAL FILLED TEFLON
COMPOUNDS^(4, 5)

Unfilled TFE	0.12
15% glass fiber	0.14
25% glass fiber	0.16
15% graphite	0.12
60% bronze	0.14
20% glass, 5% graphite	0.16
15% glass, 5% MoS ₂	0.14
Rulon A	0.21
Rulon B	0.13
Rulon D	0.15
Rulon F	0.16
15% MoS ₂	0.18
61% MoS ₂	0.28

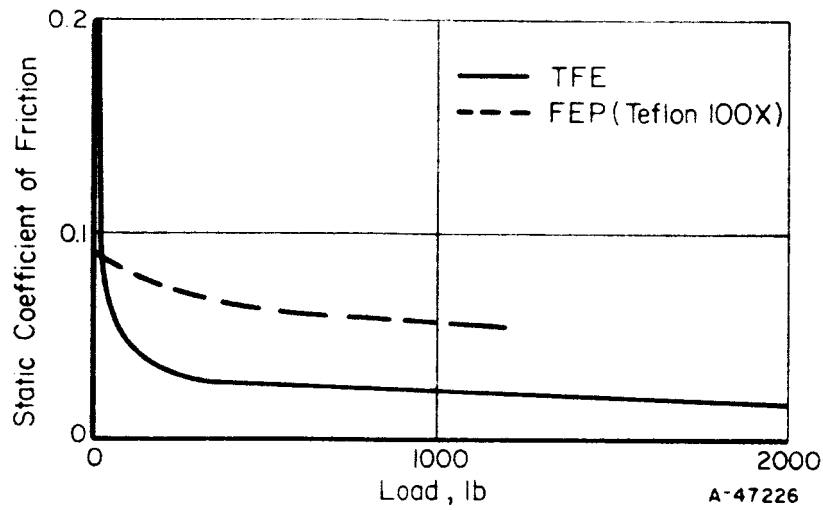


FIGURE 1. STATIC COEFFICIENT OF FRICTION OF TFE AND FEP AT 73 F^{(8)*}

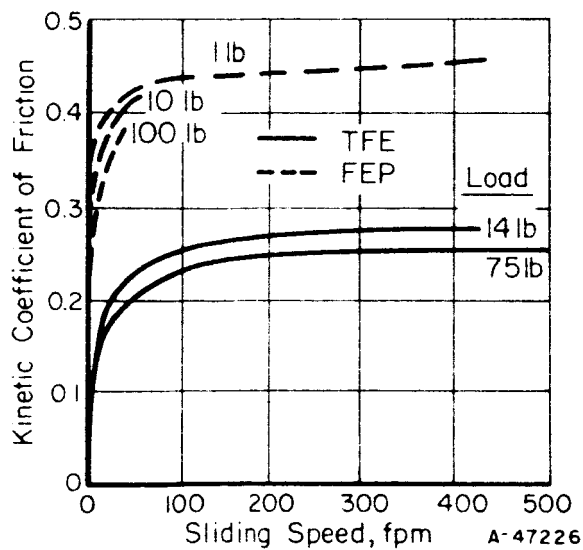


FIGURE 2. KINETIC COEFFICIENT OF FRICTION OF TFE AND FEP at 73 F⁽⁸⁾

TABLE 3. PROPERTIES OF TFE FLUOROCARBONS⁽⁷⁾Reprinted, with permission, from Machine Design

<u>Physical</u>	
Specific Gravity	2.14 to 2.19
Melting Point, °F	
Water Absorption, %	none
<u>Mechanical</u>	
Tensile Strength, 73 F, psi	2500 to 6000
Elongation, 73 F, %	250 to 400
Flexural Modulus, 73 F, psi	50,000 to 90,000
Impact Strength, 73 F	
Izod, ft-lb/in.	2.9
Tensile, ft-lb/cu in.	320
Impact Strength, -65 F	
Izod, ft-lb/in.	2.3
Tensile, ft-lb/cu in.	105
Fatigue Resistance (Cycles to Failure)	
1000-psi stress	20 million
1400-psi stress	7 million
1450-psi stress	12,000
1500-psi stress	1200
Durometer Hardness	D52
Rockwell Hardness	R58
Abrasion Resistance, Weight Loss, gm/sq in.	0.337
ASTMD1242-56 test, mg	0.35
Taber abrasion test, 10 cycles	2.2
100 cycles	8.9
1000 cycles	
Deformation Under Load, 75 F, Load = 1000 psi for 24 hr (per cent)	2.4
<u>Electrical</u>	
Dielectric Strength, Short Time, 10 mil (v per mil)	1000 to 2000
Dielectric Constant, 60 to 10 ⁹ cycles	2.1
Dissipation Factor, 60 to 10 ⁹ cycles	0.0003
Volume Resistivity, -40 to 440 F, ohm-cm	10 ¹⁸
Surface Resistivity, -40 to 440 F, ohm/sq	10 ¹⁸
<u>Thermal</u>	
Coefficient of Linear Thermal Expansion, per °F, 73 F	10 ¹⁸ 5.5 x 10 ⁻⁵
Thermal Conductivity, Btu/hr/sq ft/°F/in.	1.7
Specific Heat, Btu/lb/°F	0.25
Continuous-Service Temperature, °F	500
Heat-Distortion Temperature, °F	
66-psi stress	252
264-psi stress	132

TABLE 4. TEFLON-FLUOROCARBON RESINS⁽⁸⁾Reprinted, with permission, from Machine Design

Resin	Usual Method of Processing	Description
TFE		
Teflon 1	Molded by preforming-sintering technique; ram extruded	General-purpose powder for molding and extrusion
Teflon 5	Molded by preforming-sintering technique; ram extruded	Special molding powder - granulated for molding cylinders for skived tape
Teflon 6	Lead-press-type extrusion	Special-purpose powder for compounding and for use in extrusion
Teflon 7	Molded by preforming-sintering technique; ram extruded	Special molding powder - granulated for use where highest quality, void-free moldings are required
Teflon 30	Dip coating	Aqueous dispersion
Teflon 41BX	Compounding	Aqueous dispersion

The tendency of solid Teflon to cold flow is not considered to be a problem for this seal application. This is especially true if the cartridge design is used because most of the deformation occurs within 12 hours for temperatures from 25 to 200 C at compressive loads up to 1000 psi. (10)

Another Teflon characteristic which influences the design of a seal cartridge is the possibility that Teflon, upon contact with a mating surface, will transfer material to the mating surface, thus forming a Teflon-to-Teflon seal and giving the lowest possible coefficient of friction. (3, 1, 11) Because of this transfer phenomenon, Teflon to metal is as good as, if not better than, Teflon to Teflon. It is reported that the lowest coefficient of friction is obtained with uniform thin films on the hardest possible backing. (1)

SINGLE-SEAL INTERFACE EXPERIMENTAL STUDY

Because very little data were available for the coefficient of friction of Teflon versus a mating metallic surface in the range of loads and speeds of concern for the gimbal seal, it was necessary to determine experimentally the optimum seal interface conditions. The following sections discuss the experimental equipment and procedure used to investigate seal parameters of width, mating ring surface characteristics, Teflon type, Teflon surface finish, loading, and configuration.

Single seals were tested to reduce the number of samples which had to be fabricated. It was postulated and subsequently verified that the results of single-seal experiments would be directly applicable to the proposed design incorporating a double seal.

Experimental Equipment

The single-seal test fixture is shown in Figure 3. The basic design consisted of an inner shaft supported on a ball bearing which rotated within a housing supported on a second ball bearing. The Teflon seal was supported on a stainless steel bellows held in place by a floating plug. The plug was prevented from rotating by a pin fitted into the plug and restrained by a slot in the housing. A cap screw at the bottom of the housing provided the sealing force by acting through the plug and compressing the seal bellows. The seal plate fastened to the rotating shaft restrained the upward seal force through the upper bearing and eventually the housing.

Experimental Procedure

To compare the various seal parameters it was necessary to vary the seal load or contact stress and determine the torque and leakage at a constant air pressure. Air was admitted from a rotometer at 15 psi (the pressure difference expected in the final gimbal system) and the adjustment screw was rotated to bring the Teflon seal in contact with the seal plate while the torque and leakage were measured. The bellows deflection was measured by means of dial indicators resting on the floating plug and on the upper bellows end fitting.

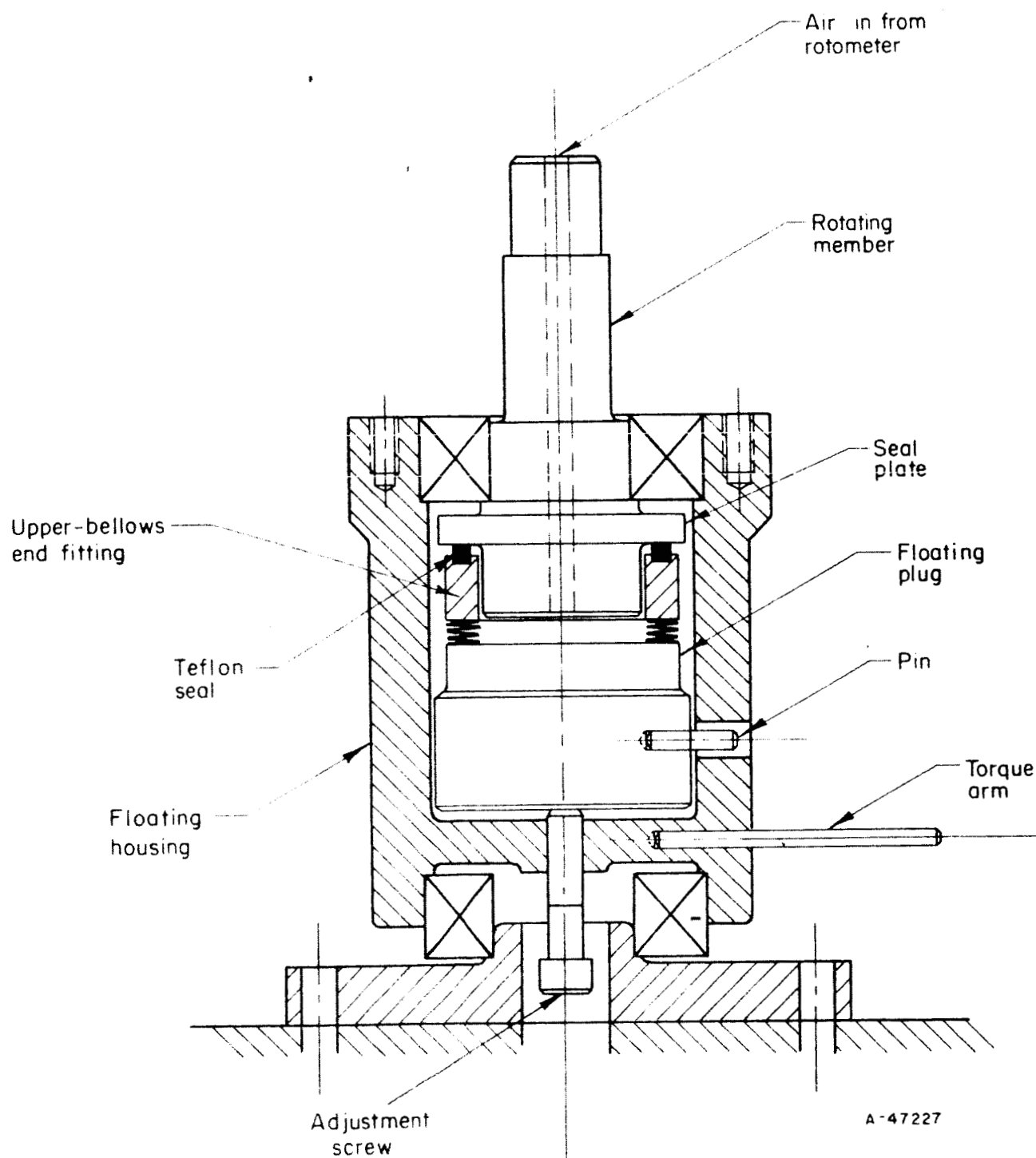


FIGURE 3. SINGLE-SEAL TEST FIXTURE

Seal torque readings were made with a torque arm and force gage attached to the outer housing. The net seal load was determined by the product of the net bellows deflection under load and the bellows spring constant. Corrections were necessary because the overbalance or underbalance of the seal being tested affected the seal loading. Calculations were based on a linear pressure distribution across the seal face using the mean diameters of the bellows and seal. Seal leakage was determined by measuring the flow rate of the inlet seal air using a rotometer and the necessary correction factors for pressure and temperature changes. The rotometers and seal bellows were calibrated prior to testing.

Seal Parameters

A series of experiments was conducted to determine the effect of the major seal variables of seal width, mating-ring surface finish, Teflon type, and Teflon surface finish on torque and leakage. Because of the number of variables and the interrelationship between leakage and torque, it was impossible to run all possible combinations of all the variables. Rather, an initial choice of seal characteristics was made on the basis of the literature review, and subsequent experiments were based on the results of the preceding ones.

The basic seal dimensions of outside diameter and inside diameter were limited because of the decision to utilize commercially available bellows in order to minimize costs and delivery times. Because the effective bellows diameter was 1.437 in., all of the seals tested had mean diameters of approximately 1-7/16 in.

Table 5 is a summary of the single-seal experiments. The data in Table 5 were taken from plotted curves. For example, the data for Run No. 7 in Table 5 were taken from the curves of leakage and torque-versus-seal load shown in Figure 4. Figure 4 is a plot of leakage and torque versus seal load for a concentric-ground ring mated with an overbalanced 1/8-in.-wide Teflon seal. The seal surface was "finished" by being mechanically pressed against a surface plate. The "knee" of the leakage curve occurs at a seal load of 2.5 lb with a corresponding leakage and torque of 43 cc per min and 3.5 oz-in. respectively. The minimum air leakage obtained was 28 cc per min at sealing loads greater than 3.5 lb. A probable operating range, although somewhat arbitrary, is included in Table 5 to illustrate the possibilities of the seal. For Run 7 the probable operating range would be a seal load ranging from 3 to 5 lb with a corresponding leakage variation of from 28 to 35 cc per min and a torque variation of 3.8 to 5.4 oz-in. Also shown in Figure 4 is a calculated torque curve based on an assumed coefficient of friction of 0.10.

Seal Width

The first group of experiments (through Run 28) indicated that, from the standpoints of torque and leakage, a seal width of 1/8 in. was better than widths of 3/16 and 1/4 in., and as good as, if not better than, a seal width of 1/16 in. A seal width of 1/8 in. was thus selected.

One seal, because of machining inaccuracies, appeared to have a contact width of approximately 0.030 in. This seal (Run 37, 38, and 39) exhibited surprisingly low

TABLE 5. SUMMARY OF EXPERIMENTAL RESULTS ON SEAL INTERFACE

Run No.	Ring(s)	Seal(s)	Load at Break	Operating Range		Minimum Leak at Load, cc/min at lb	Probable Operating Range			
			Leakage Curve, lb	Load, lb	Torque, oz-in.		Leak, cc/min	Torque, oz-in.	Load, lb	
1-6	O-ring problems, not good data									
7	CG 4	1/8 CB P	3	2.5	42	3.5	28 at 3.5	28-35	3.8-5.4	3-5
	(see Figure 5)									
8	TG	1/8 OB P	4	3.4	55	4.4	28 at 4.6	28-40	5-7.2	4-6
9	TG	1/16 OB M	13	6.5	87	7	40 at 13	--	--	--
	(too high - not acceptable) possible scratch on seal									
10	CG 1	1/16 OB M	11	8.2	80	6.5	35 at 12	--	--	--
11	CG 1	1/8 OB M - did not seal								
12	CG 1	1/8 OB M - did not seal								
13	Lap	1/8 OB M - did not seal								
14	Lap	1/8 OB P	6	5	65	5	45 at 7	45-60	5-7	5-7
15	Lap	1/16 OB M	11	8.8	80	6.2	40 at 11			
	(possible scratch on seal)									
16	Pol	1/16 OB M	10	8.1	125	10	65 at 10.5			
	(high torque - possible scratch on seal)									
17	Pol	1/16 OB M	--	13.2	185	14.8				
	(high torque - possible scratch on seal)									
18	Pol	1/16 B M	6	6	45	3.6	35 at 7	35-45	3.8-8.2	6-8
19	CG 1	1/16 OB M	4	5	42	3.4	35 at 6	35-42	3.4-6.4	5-7
20	CG 1	1/16 B M	--	8.2	65	7.4				
	(high leakage - 25 at 5, no knee)									
21	Pol	1/16 B M	--	11	130	10.4				
	(high leakage - 50 at 11.5, no knee)									
22	Pol	1/16 B P - did not seal								
23	CG 1	1/16 B P - did not seal								
24	CG 1	1/16 OB M	6-9	6.5	70	5.6	35 at 10	40-60	5.8-6.8	7-9
	(flat torque curve)									
25	Pol	1/16 OB M	4-6	4.5	60	4.8	35 at 8	40-70	4.2-7	4-7
26	CG 1	1/8 OB P	5-6	5.8	87	7	5 at 6.5	10-70	7.2-10.2	4-6
	(high torque)									
27	CG 1	1/8 OB P	5-9	6.6	85	6.5	5 at 9			
	(leakage hysteresis - high torque)									
28	CG 1	1/8 OB P	5	2.5	45	3.6	5 at 7	5-35	4-5.4	3-5
	(leakage hysteresis - high torque)									
29	CG 1	1/16 TC - wouldn't seal								
30	CG 1	1/8 TC - wouldn't seal								
31	CG 1	1/8 OB P	--	10	85	6.8				
	(almost vertical leak and torque curves)									
32	CG 1	1/16 B P	--	6.1	45	4.1				
	(leakage hysteresis)									
33	CG 1	1/8 B P	6-8	6.6	40	3.2	5 at 8.5	25-75	2-4.6	6-7
	(almost vertical curves - probably distressed)									
34	CG 1	1/4 P 15 - did not seal, vertical torque curve								
35	CG 1	1/18 OB P	--	8.2	35	6	5 at 11.4			
	(extreme leakage hysteresis)									
36	CG 1	B 128 - P P - wouldn't seal								
37	CG 1	C 18 OB P	--	7	20	1.4	5 at 6	7-17	1.4-2.2	2-4
	(hysteresis of both leakage and torque - minimum values)									
38	CG 2	C 18 OB P	2-3	8	30	2.4	45 at 5.5	15-50	1.6-3.6	2-4
39	CG 3	C 18 OB P	--	7.1	20	1.6	45 at 5.5	10-27	0.8-2.6	1-3
40	CG 3	B 18 OB P - wouldn't seal								
41	CG 3	A 18 OB P	--	4.1	80	6.4	35 at 5.5	35-65	6.4-9.6	4-6
	(right figure)									
42	CG 2	A 18 OB P	--	8.5	80	7	25 at 10.7 - leakage too high, no knee			
43	CG 2	B 18 OB P	--	8.2	30	2.4	5 at 10.1 - leakage too high, no knee			
44	CG 3	B 18 OB P	--	4.7	25	2.8	5 at 10.5 - leakage too high, no knee			
45	CG 3	C 18 OB P	2-4	7	20	1.8	45 at 5	10-25	1.2-2.4	2-4
	(at 10 psi or less)									
46	CG 3	C 18 OB P	2-4	7.5	15	1.5	45 at 4	5-22	0.6-1.8	2-4
	(at 2 psi or less)									
47	CG 3	1/16 OB TC 1 - wouldn't seal								
48	CG 3	1/8 B TC 1 - wouldn't seal								

TABLE 1. Test Conditions

Run No.	Ring (a)	Seal (b)	Load at Knee of Leakage Curve, lb	Pressures			Maximum Leak at Load, cc/min at lb	Probable Operating Range		
				Load, lb	Leak, cc/min	Torque, oz-in.		Leak, cc/min	Torque, oz-in.	Load, lb
44	CG 3	C 1.8 OB P	4-5	4.6	45	2.4	<5 at >7.5	12-45	2-3.8	4-6
50	CG 3	C 1.8 OB P	3-5	4.1	45	3.6	<5 at >7.5	25-55	2-7.2	3.5-5.5
51	CG 3	C 1.8 OB P	Fluctuation in flow	probably dirty seal, not good data						
52	CG 3	B 1.5 OB P	7-11	7.8	75	6	<5 at >7.5	10-45	1.8-5.0	2-4
(torque and leakage curves almost vertical)										
53	CG 3	E 1.8 OB P	2-4	3.1	30	2.5	<5 at >5.5	10-32	2.4-3.4	3-5
54	CG 3	F 1.8 OB P	2-5	3.1	40	3.4	<5 at >7	15-45	3.2-5.0	3-5
55	CG 3	F 1.8 OB P	3-5	2.9	45	3.5	<5 at >5.4	7-45	3.8-5.2	3-5
56	CG 3	F 1/8 OB P	<5	--	--	--	<5 at >4.7	5-15	4.6-6.6	3-5
(no data below 3-lb load)										
57	CG 3	F 1/8 OB P	2	2.5	35	2.8	<5 at >5.8	10-45	1.8-5.0	2-4
58	CG 3	G 1.8 OB P	1-3	1.5	35	2.8	<5 at >3.5	12-35	2.8-5.5	1.5-3
59	CG 3	G 1.8 OB P	1-3	1.4	32	2.6	<5 at >3.3	7-30	2.8-4.9	1.5-3
60A	CG 3	E 1.8 OB P	1-5	2.7	52	4.2	<5 at >7.7	30-70	3.8-5.1	2-4
60B	CG 3	F 1.8 OB P	1-5	2.1	50	4.4	<5 at >6.5	27-75	4.0-5.6	1.5-3.5

(a) "Ring" Column

CG = concentric ground

IG = table ground

Lap = lapped surface

P = polished surface

(b) "Seal" Column

B = Teflon face pressed against surface plate with a load corresponding to a contact stress of 1000 psi.

OB = Overbalanced design - an increase in internal pressure results in an increase in seal load

M = Teflon face in as-machined condition

UB = Underbalanced design - an increase in internal pressure results in a decrease in seal load

B = Balance design - seal load independent of internal pressure

FC = Teflon coating on metal substrate, coated by spraying with water suspended and fused at 700 F, one coat, approximately .004 mil thickness

FCS = Teflon coated with eight coats of FC, suspended and approximately .004 mil thickness

A, B, C, etc. = specimen designation

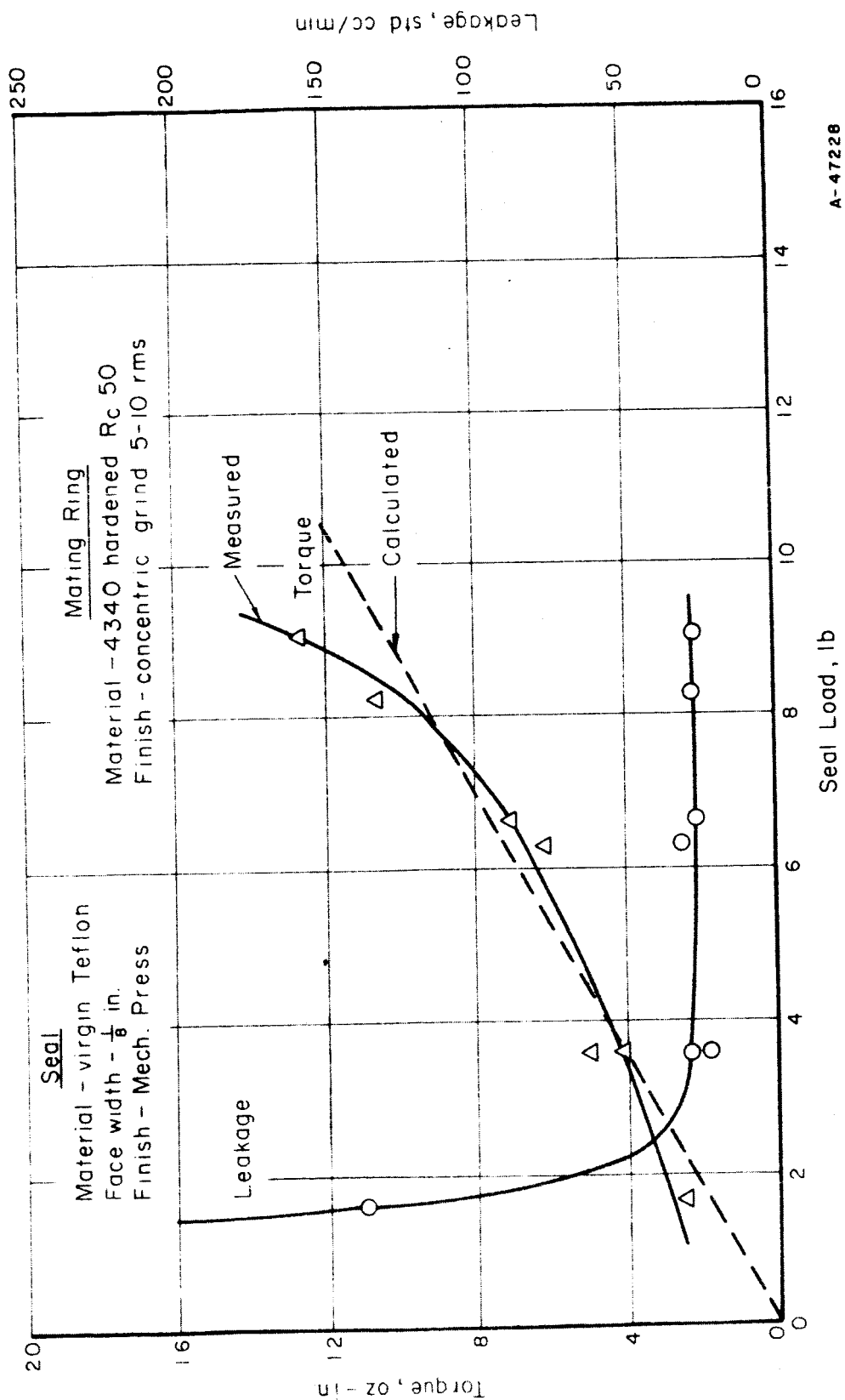


FIGURE 4. LEAKAGE AND TORQUE VS. SEAL LOAD

torque and leakage values. It seems reasonable to assume that the corresponding torque and leakage values for pressures of 10, 15, 20, and 25 lb per sq in. are 27 cc per min respectively. The possibility of determining the seal width for pressures of 10 - 25 lb per sq in. by means of very narrow seal widths is further indicated by the fact that the narrow, flat-faced seal, run at 25 lb per sq in. contact with the mating ring. The low values obtained in runs 27-30 could be due to the fact that an attempt to obtain an ultra-low torque and leakage value were discontinued. However, the possibility exists that an ultra-low configuration might exist for the low values obtained.

Mating-Ring Surface Finish

Four types of surface finish on the hardened-steel mating ring were investigated briefly:

- (1) A longitudinally ground surface.
- (2) A concentrically ground surface.
- (3) A 600-grit lapped surface.
- (4) A polished surface.

Theoretically, the concentrically ground surface was the most attractive because of the tendency of the tool marks to minimize leakage and torque. Preliminary tests indicated that there was not a great difference between the four types of surfaces, although there appeared to be some difficulty in consistent sealing if the surface was too smooth. The decision was made to complete the program with hardened (R_C 59) 4540 steel rings concentrically ground to a surface finish of 5 to 10 microinches rms by means of a D more grinder-lathe attachment. No attempt was made to specify or achieve a particular flatness. Because satisfactory results were obtained, this type of mating-ring surface finish is recommended.

However, in view of the tentativeness of the original tests, and in view of the transfer of Teflon to the mating ring surface, it is quite possible that other types of mating-ring surface finish would function as well.

Teflon Type

The advantages of virgin TFE fluorocarbon resins as opposed to filled compounds of FFE fluorocarbon resins were discussed previously. All of the single-seal experiments were performed with the TFE fluorocarbon resin designated as Teflon 1 with the exception of four metal specimens which were coated by means of an aqueous dispersion of TFE fluorocarbon resin designated as Teflon 11BX (Runs 26, 30, 47, and 48). Both 3/4-in. and 5/8-in. outer diam. and 1/4-in. and 3/8-in. seal widths were investigated and extremely poor results were obtained, which may be due to the difficulty of obtaining a smooth coating or it may be a function of the type of Teflon. It may be possible to use a Teflon coating if the Teflon surface is machined and pressed. However, the advantages of a Teflon coating may appear to be sufficient to warrant a more detailed investigation.

Because all of the seals were constructed from a single rod of Teflon, material variability is not a problem and thus direct comparisons of seal performance were possible without a detailed investigation of the variability of Teflon materials. However,

due to the possible variability of Teflon material when obtained from various suppliers or even different batches from the same supplier, it is recommended that a more consistent grade of TFE, such as Teflon 7, be considered for the eventual seal.

Teflon Surface Finish

Unsuccessful attempts were made to determine the surface finish and flatness of the Teflon specimens by means of a profilometer (the diamond stylus gouged the surface) and a monochromatic light (the interference bands did not show up against the white Teflon surface). Discussions with various Teflon manufacturers and users specializing in machining Teflon to close tolerances also uncovered no practical methods of surface measurement. A significant improvement in seal performance was obtained, however, by cold pressing the Teflon against an optically flat surface plate. This cold-pressing process gave consistently better results than the machined and the machined-and-polished surface finishes. This can be seen by comparing the results of Runs 11, 12, and 13 with the results of Runs 7, 8, and 14. It also appears that, even though impossible to measure at present, the surface finish of the Teflon may be of more significance than the surface finish of the mating ring. Consequently the production process for the Teflon surface must be carefully controlled. This process is described in detail in the following paragraphs.

Cold-Press Procedure. The Teflon is first machined roughly to size leaving stock for final machining on both the seal face and the sides of the seal (surfaces A, B, and C in Figure 5). The Teflon is then pressed into the metal carrier which is machined to

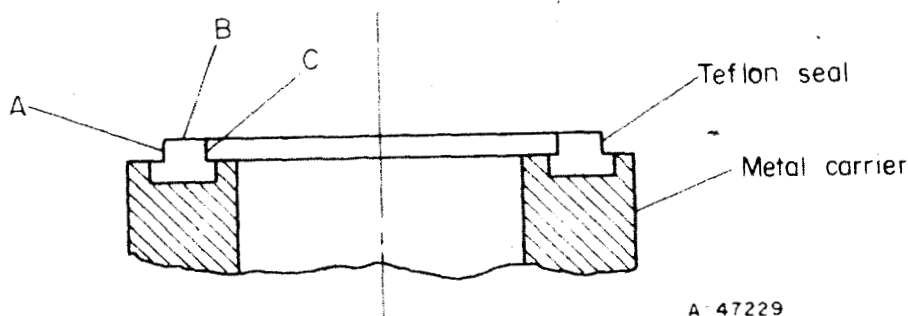


FIGURE 5. TEFLON SEAL.

provide a light interference fit with the Teflon. The insertion load should be approximately 10 or 15 per cent greater than the eventual load which will be used to press the seal against the surface plate. This ensures that the second loading will not disturb the orientation of the Teflon in the metal carrier.

The final machining is then done with a high-rake tool bit which has been dressed on a diamond wheel to a $3/32$ -in. radius. The high rake is to prevent Teflon "pile-up" on the tool while the diamond wheel is to give the tip of the tool a smooth finish. The final cut on the seal face (surface B) should be a light cut with a low feed. For a seal

With an average diameter of about 1-3/8 in., typical lathe settings have been: feed, 0.001 in. per revolution; speed, 145 rpm (215 ft/min surface speed); depth of cut, 0.002 to 0.005 in. The edges of the seal and mating surface are then machined by feeding the tool toward the metal carrier to avoid leaving a burr on the edge of the seal face.

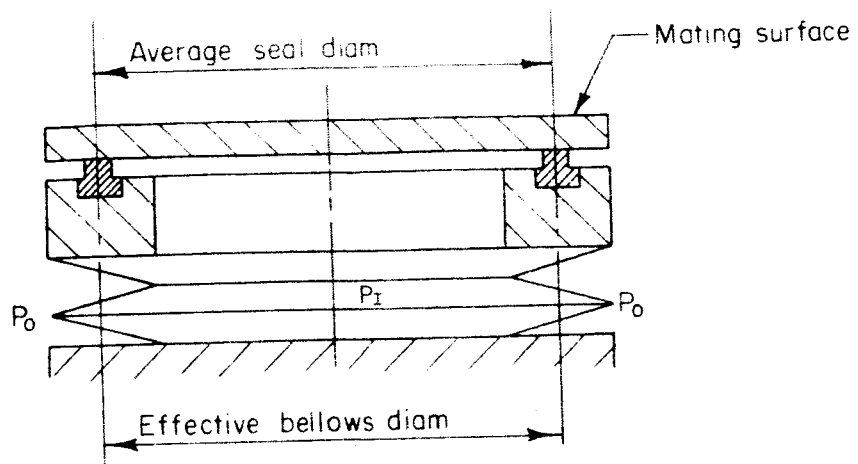
The final step is to press the seal face (surface B) against an optically flat metal surface plate with a load sufficient to cause a contact stress of 2000 psi. The load should be held for 5 minutes to allow time for the Teflon to "flow". For a seal area of 0.539 sq in., this load would be 1078 lb. A load of 1200 lb would then be a typical value for the initial load to seat the Teflon in the groove. Visual inspection of the Teflon transfer to the surface plate is helpful in determining whether or not a seal "takes a press". A continuous uniform Teflon "smear" should be noticeable on the surface plate.

Balance Mode

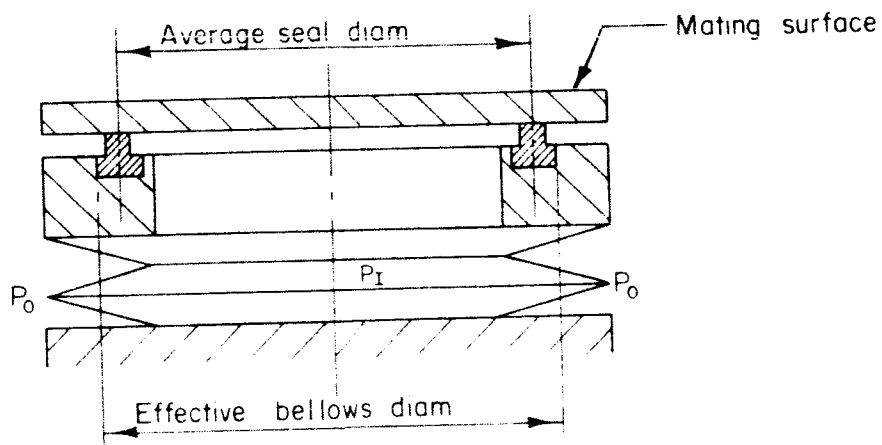
Balance is a term applied to mechanical face seals which describes how the pressures being sealed affect the contact pressures of the mating surfaces. The balance condition of a face seal can be changed by changing the effective diameters upon which the external and internal pressures act. This can be seen in the bellows/seal configuration shown in Figure 6. If the effective seal diameter and the effective bellows diameter are the same (Figure 6a), then a change in the internal pressure, P_i , will have no effect on the contact pressure of the seal and the mating surface. If the effective bellows diameter is larger than the effective seal diameter (Figure 6b), then an increase in the internal pressure, P_i , will cause an increase in the contact pressure between the seal and the mating surface. Likewise, if the effective bellows diameter is smaller than the effective seal diameter (Figure 6c), an increase in the internal pressure will cause a decrease in the contact pressure between the seal and the mating surface. These three conditions are referred to as balanced, overbalanced, and underbalanced, respectively. The contact pressure, along with the surface characteristics of the seal and mating surface, and the resulting coefficient of friction, will determine the torque required for the seal. The contact pressure in conjunction with the surface finishes and flatness of the mating surface and seal, will determine the amount of leakage which will occur. The balance condition, then, is important when a variation in internal pressure is expected.

Theoretically an ideal face seal should be designed so that it is perfectly balanced, i.e., so that a variation in air pressure has no effect on the seal load. From a design standpoint this is impossible due to machining tolerances and a lack of knowledge concerning the pressure distribution across the seal face. If an attempt is made to design a balanced seal there is a definite possibility that the seal may shift from one mode to another and wear occurs on the surface and, thus, the pressure distribution across the seal face will change. Therefore, most seal manufacturers design for a definite overbalanced condition to insure that the pressure will affect the seal in a known manner. Most of the experimental seal designs for this reason have been overbalanced designs.

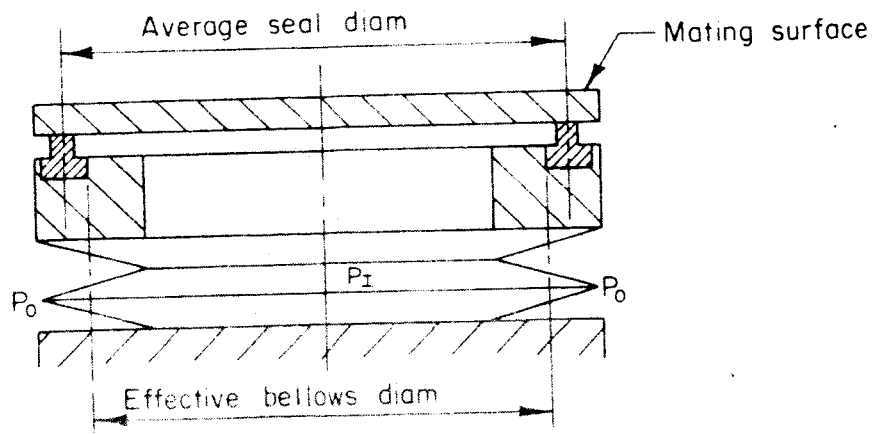
To determine more closely the effect of the overbalanced designs on torque and leakage, three experiments (Ref. Nos. 14, 15, and 16) were conducted with the same seal design, testing ring at air pressures of 1, 10, and 50 psi. The additional seal load due to the air pressure in the overbalanced design tested was 2.1, 1.4, and 0.7 lb per the 1, 10, and 50 psi pressures respectively. These experiments showed only a minor variation in torque and leakage with the variation in air pressure. The



a. Balanced Seal



b Overbalanced Seal



c. Underbalanced Seal

A-47230

FIGURE 6. SEAL CONFIGURATIONS

overbalanced design is particularly attractive in light of experiments conducted with 1/8-in. overbalanced and balanced seals. The overbalanced configuration gave approximately the same torque and leakage values as the balanced configuration although at generally lower seal loads and with more consistent results.

Selected Seal Configuration

The seal interface dimensions selected for the double-seal evaluation work are shown in Figure 7. The configuration was a 1/8-in.-wide Teflon seal of overbalanced design, with a 1.373-in. mean diameter and 0.539 sq in. area, finished by means of the cold-press procedure described above. The mating ring was concentrically ground with a 5 to 10-rms surface finish. The experimental seal dimensions were chosen to be consistent with the dimensions of standard off-the-shelf bellows to minimize costs and delivery times. Assuming that the effective seal diameter was the same as the average diameter, the dimensions shown in Figure 7 resulted in an overbalanced area of 0.141 sq in. The additional seal load caused by the air pressure was 2.12, 1.41, and 0.71 lb at pressures of 15, 10, and 5 psi, respectively. To obtain the total seal load, these values were added to the seal load caused by the bellows deflection.

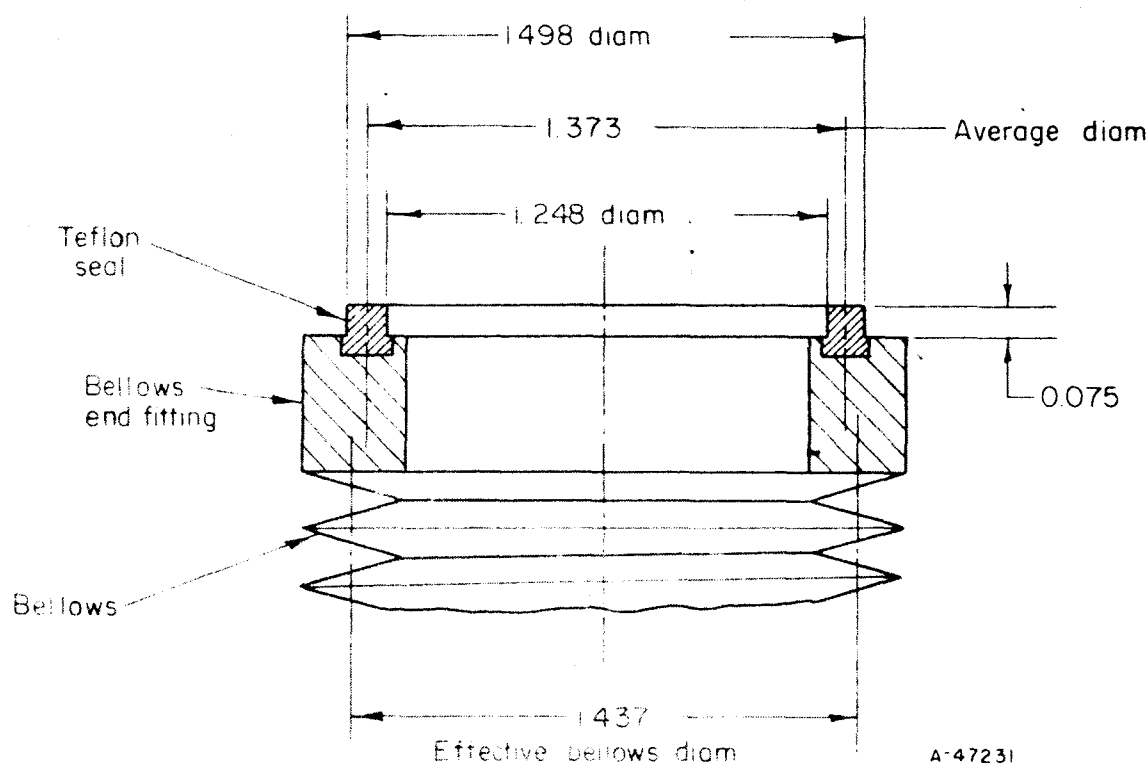


FIGURE 7. SEAL-INTERFACE DIMENSIONS

Data Reproducibility

Six experiments (6A, 6B, 6C, 6D, 6E, and 6F) were performed with the same Teflon seal and mating ring to determine the reproducibility of the data. After each run, the test fixture was completely disassembled, the Teflon seal and the mating ring were washed with methyl ethyl ketone, and the test fixture was reassembled.

The six sets of data obtained are plotted in Figure 8. The curves shown are the upper limits of torque and leakage obtained. Thus, if a seal-load range of 3 to 5 lb were considered probable, it could reasonably be expected that the torque would be within a range of from 5.7 to 7.5 oz-in. and that the leakage would be within a range of from 70 to 35 standard cc per min.

Three 1/8-in.-wide Teflon seals machined to the dimensions given in Figure 7 were tested against a single mating ring to determine the repeatability of the Teflon surface. The results of these experiments (Runs No. 53, 54, 55, 57, 58 and 59) are shown in Figure 9. Although the repeatability of the torque data was less than was hoped for, the leakage data were very encouraging.

Recommended Seal Load

To specify a seal load for a gimbal-seal cartridge design, a compromise must be made between low leakage and low torque. Assuming an upper leakage limit of 50 cc per min, the seal load must be a minimum of 3.9 lb as shown in Figure 8. Construction tolerances, errors in load measurement, etc., may account for load variations as much as 1/2 to 1 lb. Using 8/10 of a pound as a possible load variation, the seal load specified then must be $3.9 + 0.8$ or 4.7 lb. If the 4.7 lb is specified, the same tolerance could be expected on the high side - i.e., the load could be expected to vary up to $4.7 + 0.8$, or 5.5 lb. This would mean that with a specification of 4.7 lb, the torque could be expected to vary between 6.4 and 7.9 oz-in. and the leakage could be expected to vary between 50 and 30 cc per min.

DOUBLE-SEAL EXPERIMENTAL STUDY

The primary purpose of the double-seal experimental study was to evaluate the leakage and torque characteristics of a double-seal assembly as pressed against the mating rings by a common bellows, and to simulate the loading conditions expected to exist during the operation of the seal cartridge in the gimbal system.

No attempt was made to simulate the radial air passage required in the final seal cartridge, thus again allowing standard, off-the-shelf bellows to be used. The standard bellows (Sealol Inc. Part No. A-30197-18) included stainless steel end fittings which were machined to accept the Teflon seal shown in Figure 7.

In the following three sections the experimental equipment, the test procedure, and the results of the double-seal evaluation work are described.

Experimental Equipment

Figure 10 is a schematic drawing of the test fixture which was constructed to perform the double-seal interface experiments. Two problems arose which made a second test fixture necessary: (1) the dial gages used to measure the bellows deflection for the single-seal experiments caused the double seal to tilt and not contact the mating ring properly; (2) the addition of the torque arm to the bellows assembly caused the

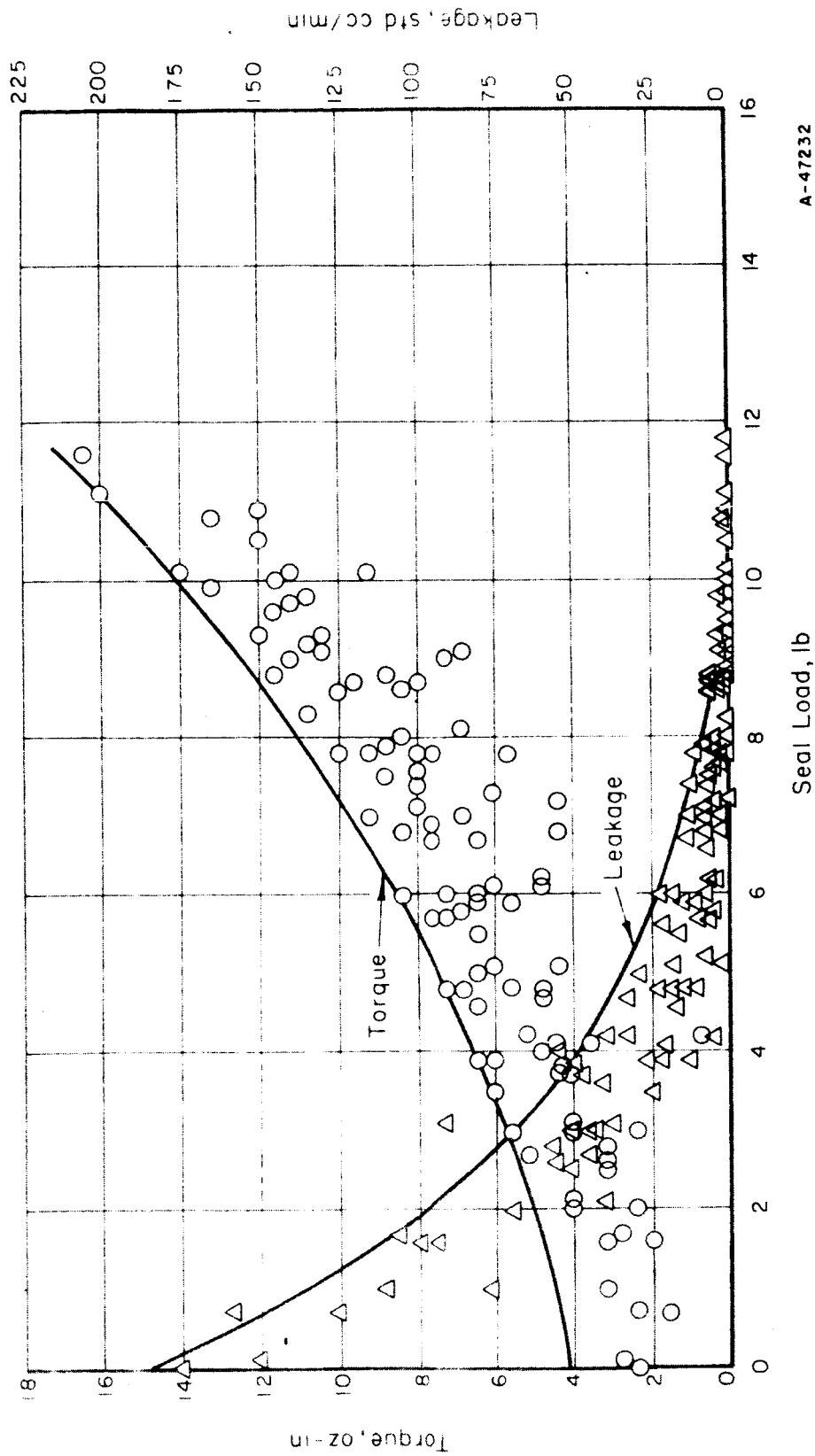


FIGURE 8. LEAKAGE AND TORQUE VS. SEAL-LOAD

Run No. 53, 54, 60A, 60B, 61A, and 61B
TFE seal E, mating ring CG3.

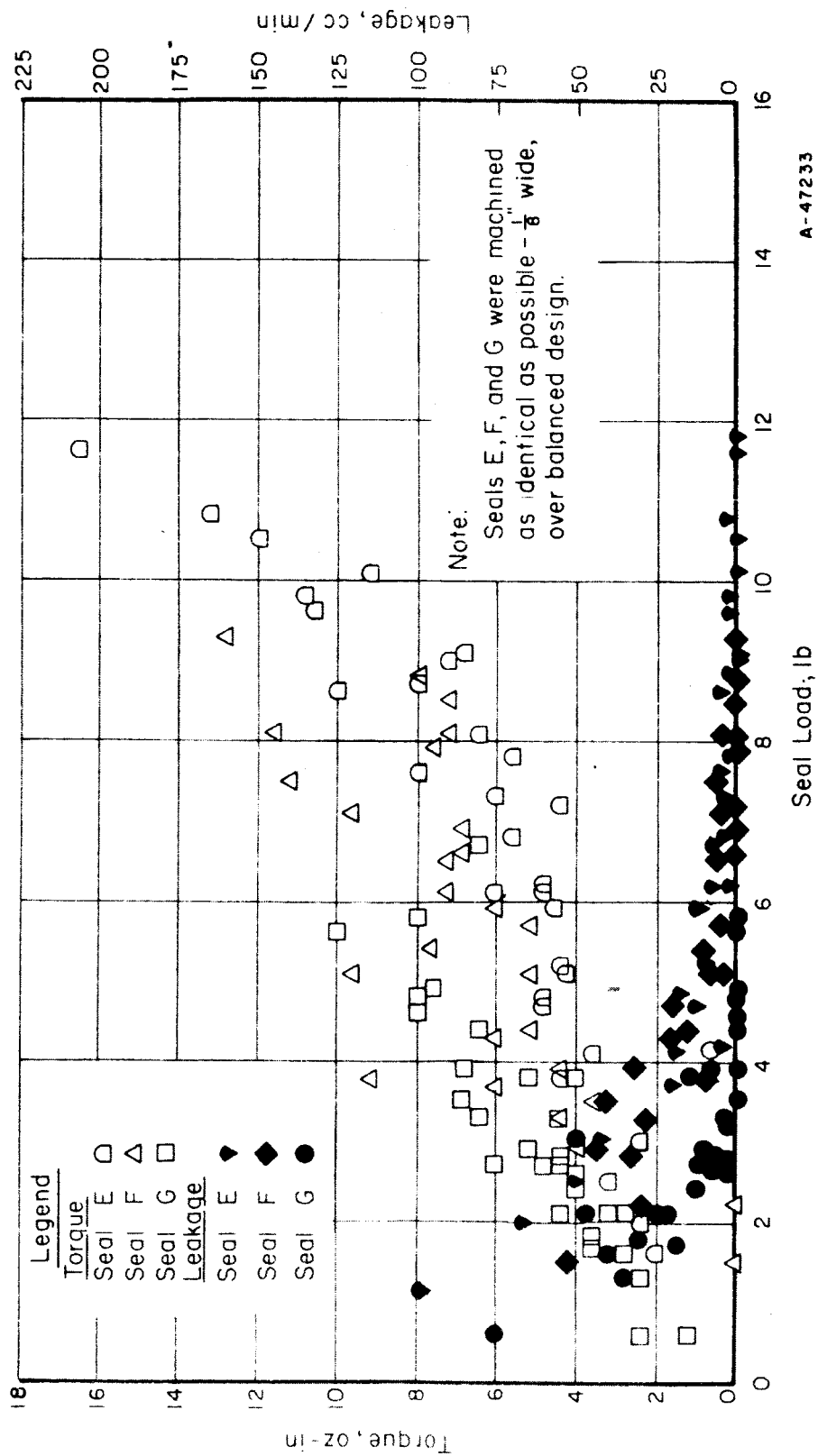
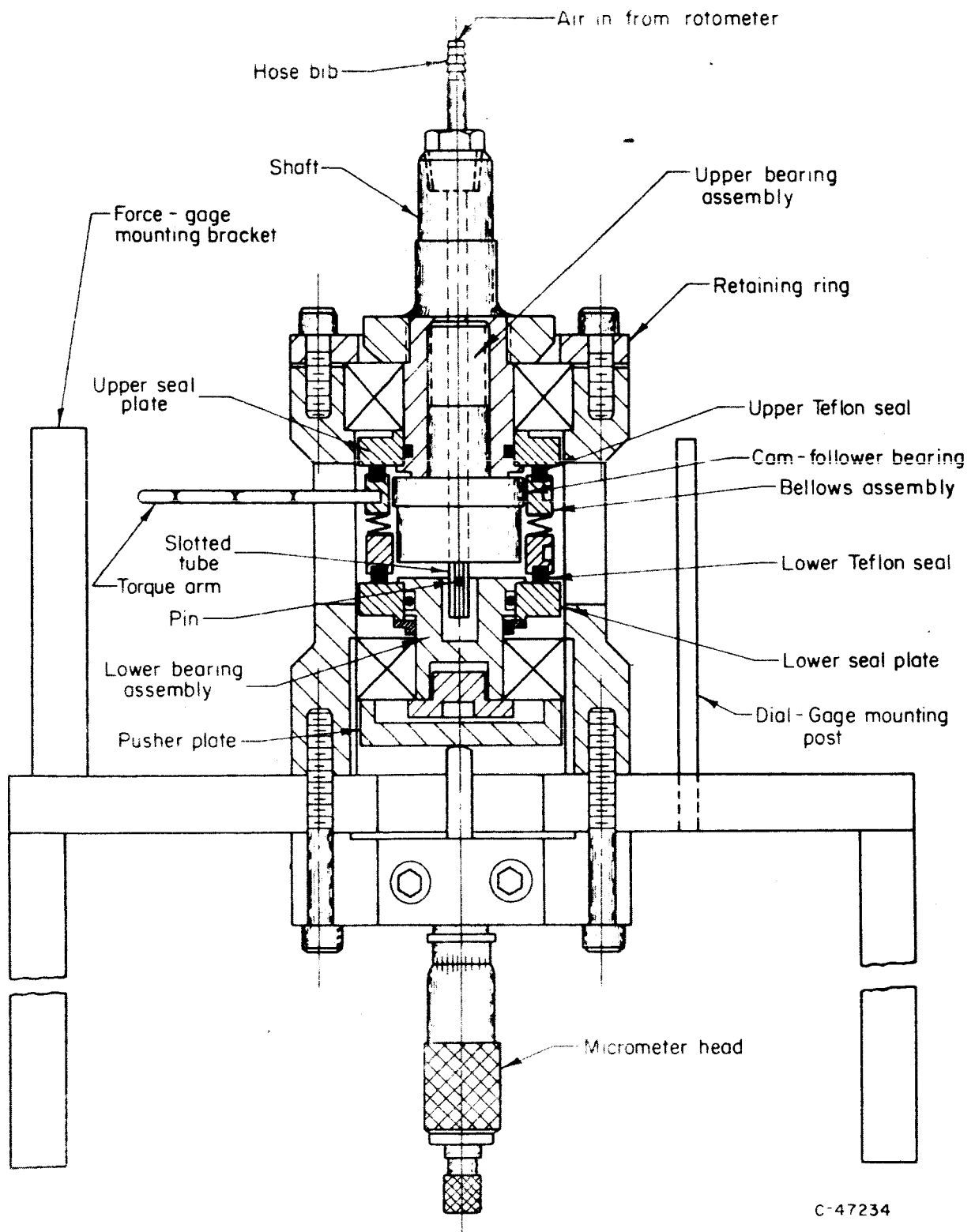


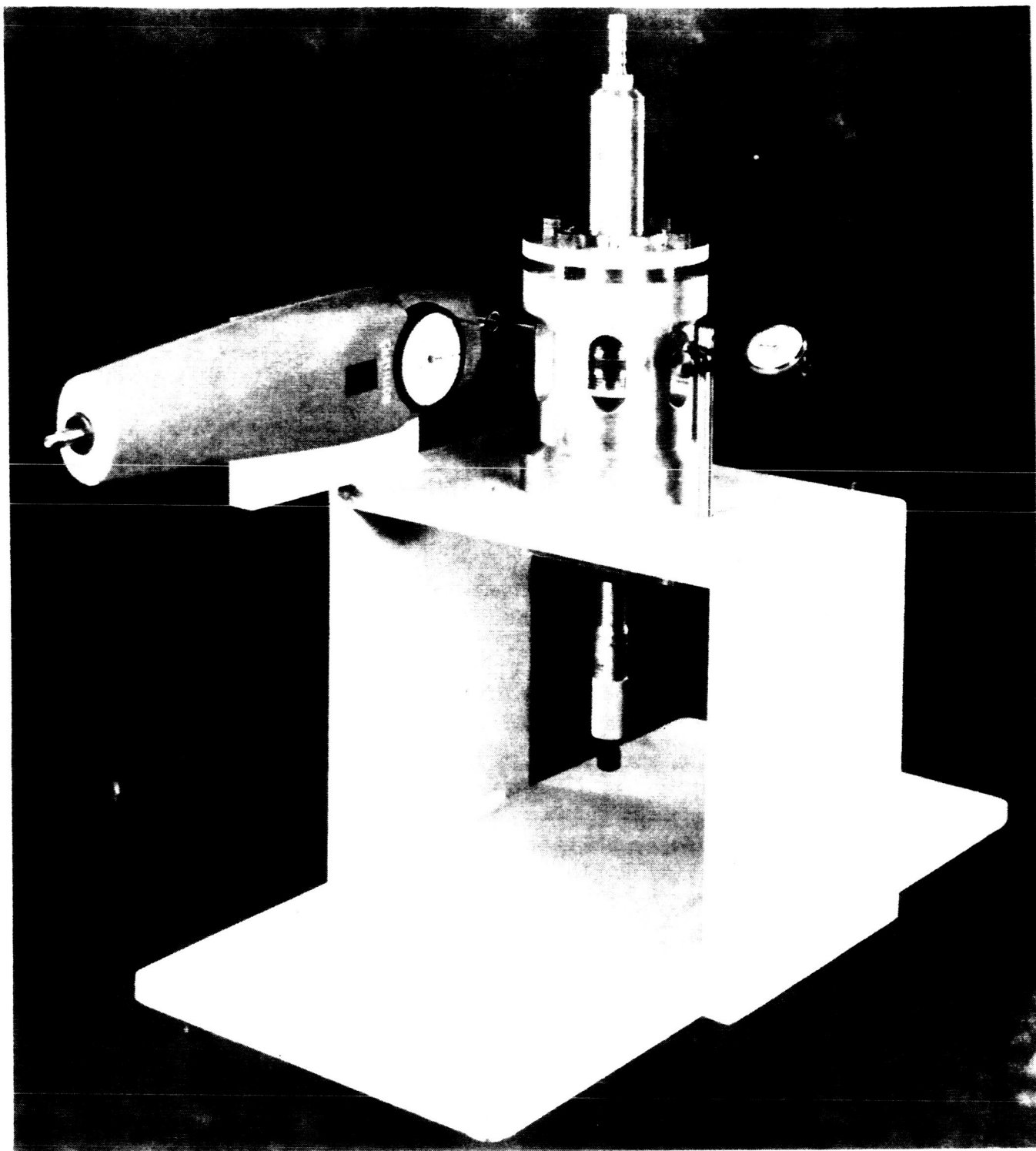
FIGURE 9. SEAL LOAD VS. TORQUE AND LEAKAGE

Run No. 53, 54, 55, 57, 58, 59.
TFE seal E, F, G; mating ring CG3.



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FIGURE 10. DOUBLE-SEAL TEST FIXTURE

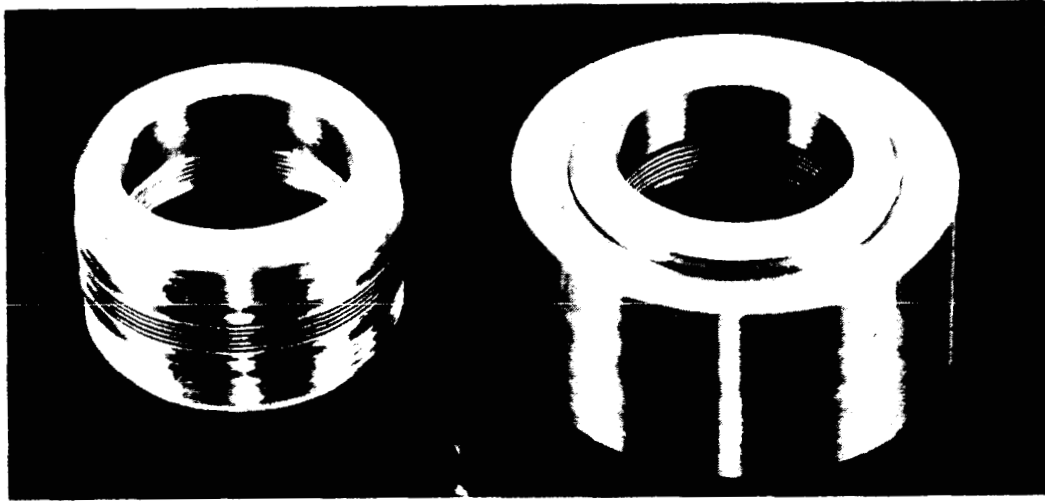


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FIGURE 11. DOUBLE-SEAL TEST FIXTURE

bellows assembly to move sideways. The new test fixture eliminated these problems by (1) using the micrometer head to measure bellows deflection, and (2) the addition of the cam follower bearing to keep the bellows assembly centered. The test fixture, including the force gage, micrometer head, and dial indicator, is shown in Figure 11.

In order to press the Teflon seal into the bellows end fitting, a seal press fixture was constructed as shown in Figure 12. The fixture was screwed onto the outside of the bellows end fitting on which threads had been cut. The seal press fixture extended down such that the bellows was not compressed when the Teflon was pressed against the surface plate.



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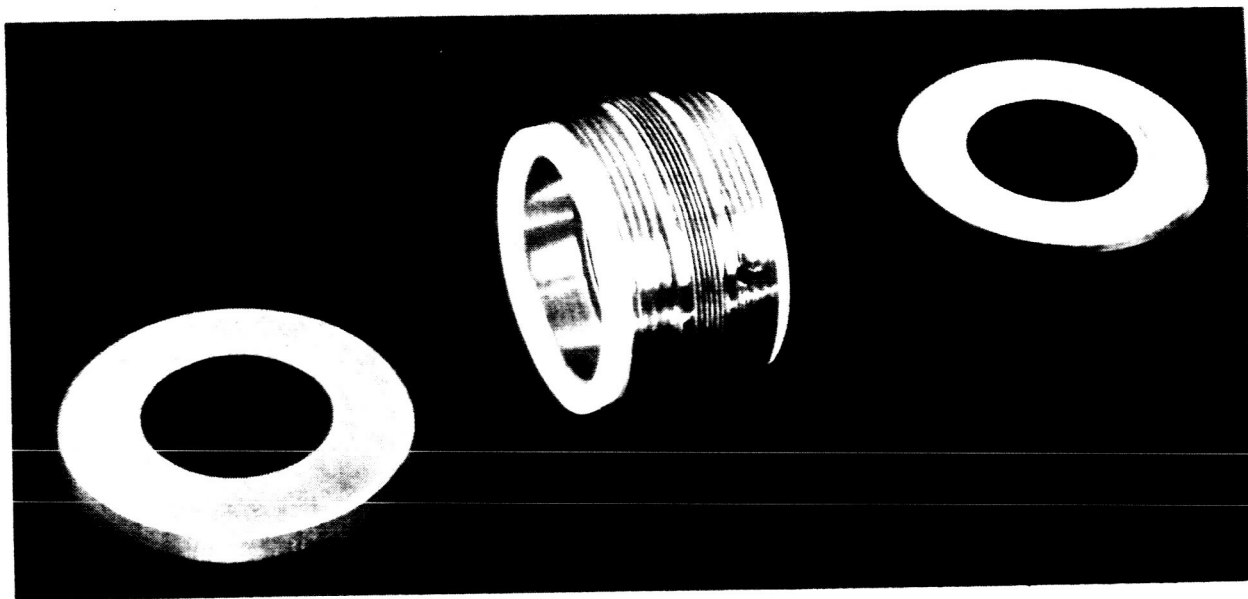
FIGURE 12. SEAL-PRESS FIXTURE

Bellows Installation for Test

A bellows assembly and seal plates are shown in Figure 13. To install a bellows assembly for test, the retaining ring (Figure 10) and the upper bearing assembly are removed from the test fixture. The upper and lower seal plates and the surfaces of the upper and lower Teflon seals are cleaned with acetone and a nonlinting-type tissue. The bearings should be lubricated with a small amount of light machine oil. Figure 14 is a photograph of the lower bearing, bellows, and upper bearing assemblies which are shown schematically in Figure 10.

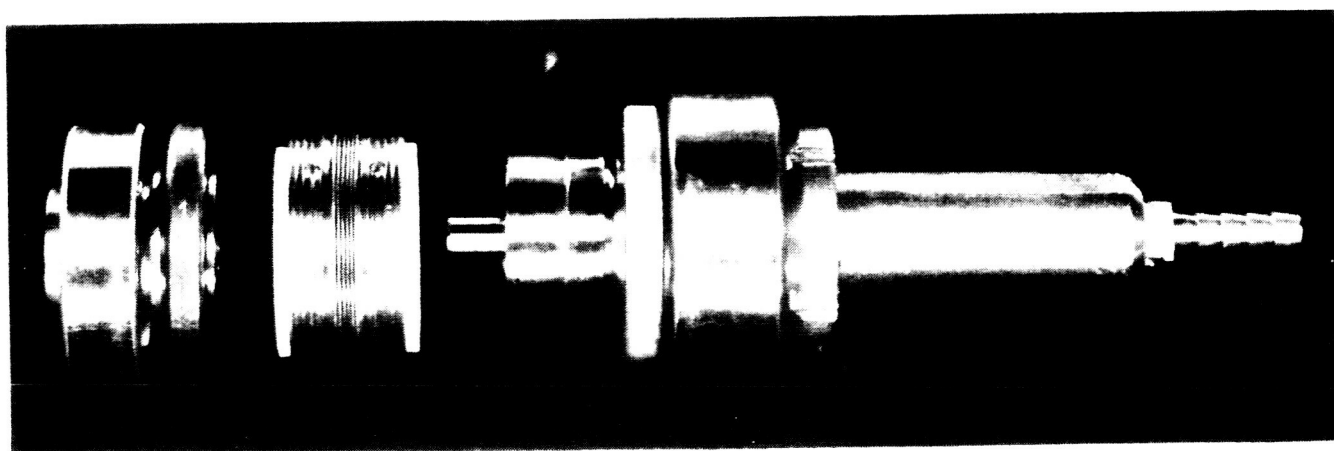
The micrometer is retracted. The bellows assembly is then set into position and the torque arm screwed in finger-tight. The torque arm should be approximately perpendicular to the force gage mounting bracket. The bellows assembly is considered to be right side up when the torque arm is in the upper position as is shown in Figure 10.

The upper bearing assembly is then positioned so that the slotted tube engages the pin in the lower bearing assembly and is clamped into place with the retaining ring. The bearing OD must be clean and free from grease and dirt or it will not fit properly into the case.



8381

FIGURE 13. BELLOWS ASSEMBLY AND SEAL PLATES



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FIGURE 14. LOWER BEARING, BELLOWS, AND UPPER BEARING ASSEMBLIES

The force gage should be mounted on the bracket in that the center line of the gage will be level with the torque arm when force readings are being taken. To fit over the torque arm, the hook must be loosened approximately 1/4 turn.

The air supply to the seal is provided by a hose connected to the hose bib. The hose should be suspended from above to provide a length to twist when the shaft is rotated. Flowmeters and the pressure gage are placed between the air pressure regulator and the test fixture. With the pressure gage on the downstream side of the flowmeter, the flowmeter should be calibrated at operating pressure. The three flowmeters connected in series shown in Figure 15 have capacities of 0 to 80, 0 to 110, and 0 to 1500 atm cc/min. These flowmeters were calibrated and used for both the single-seal and double-seal tests. The latter two would have been adequate for the double-seal tests. The hose shown in Figure 15 is not shown suspended from above as should be the case during the actual testing.

Test Procedure

The double-seal experiments were conducted utilizing a different test procedure than was used for the single-seal tests. The seal was loaded before the air pressure was applied to more closely approximate the actual conditions which will exist during the fabrication of the seal cartridge and the subsequent operation of the seal cartridge in the gimbal system.

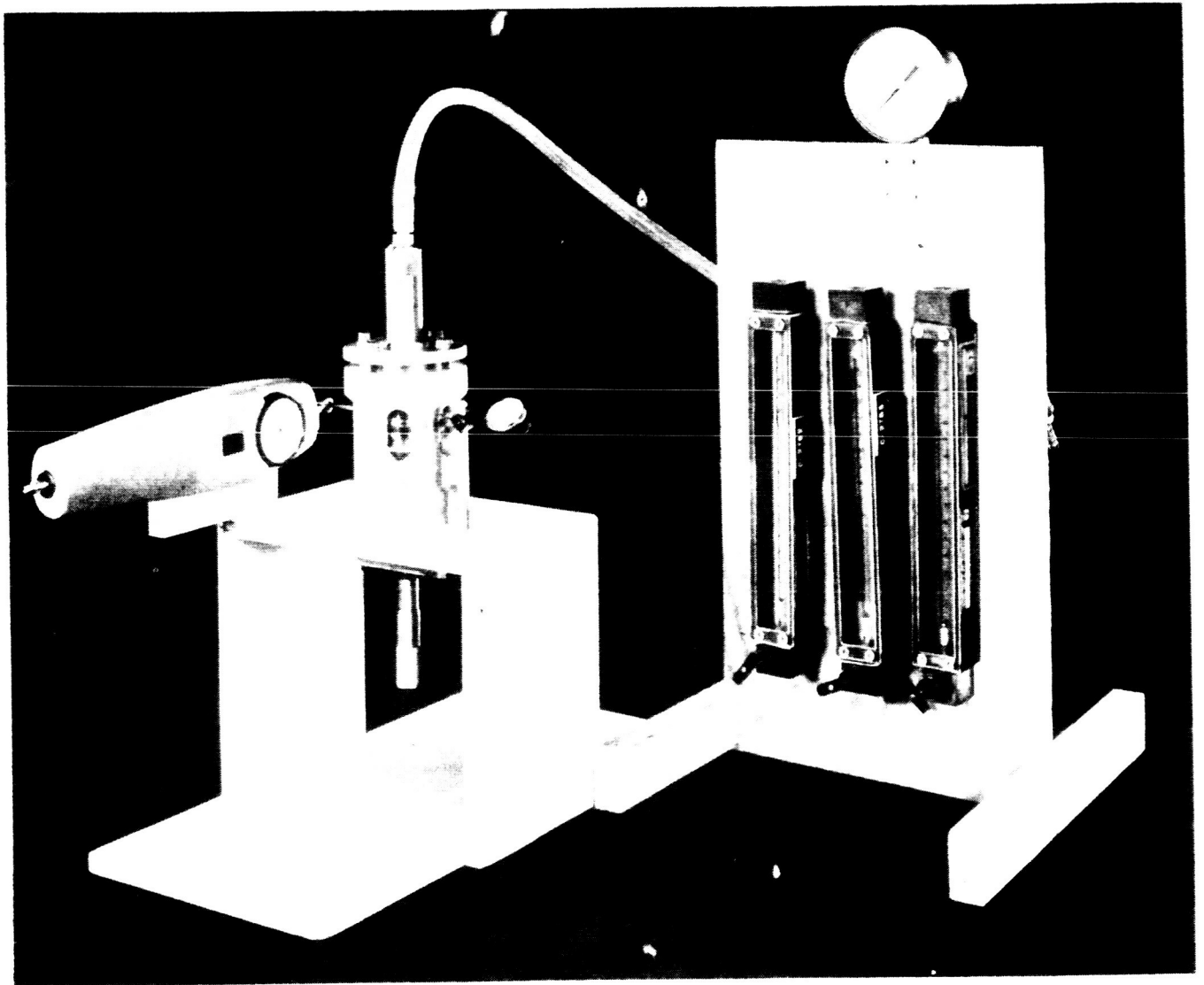
With the dial indicator on the mounting post (Figure 10), the indicator point is set into the top hole of the bellows and the bellows is slowly raised by the micrometer until the indicator stops. This is the point at which the seal makes initial contact with the seal plate. The micrometer reading at this point is the tare or zero reading. The dial indicator is removed after obtaining the zero reading.

The seal load is preset to the desired load by means of the micrometer. The spring rates of the bellows (A and B) are 170 lb. per in. This means that a net movement of 0.001 in. on the micrometer will change the seal load by 0.17 lb. The bellows should never be bottomed out because of the possibility of overstressing the bellows and hence changing the spring rate.

The air is then turned on and increased up to operating pressure. Flowmeter and force-gage readings are taken while slowly rotating the shaft in a clockwise direction by hand. Readings are not taken during the counterclockwise rotation necessary to unwind the tubing.

The outer 1/2 notch in the torque arm, which is the notch engaged by the force gage is 2-1/2 in. from the seal center line. Torque in oz-in. is therefore obtained by multiplying the force gage reading by four (40).

When a Teflon-seal and seal-plate combination is run initially it usually is necessary to "run-in" the seal. The bellows will be unstable until the seal has been "run-in". Normally one to three runs will be sufficient. During the "run-in" Teflon flakes may appear on the seal plates. The seal plates should be cleaned after each run.



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FIGURE 15. DOUBLE-SEAL EXPERIMENTAL SETUP

Double-Seal Experimental Results

The double-seal experiments were conducted with two bellows assemblies (A and B) and two seal plates (Nos. 1 and 3). Several tests were conducted at each of three different bellows deflections. The 0.015-in. deflection simulated ideal assembly conditions. This deflection provided a 4.7-lb load on the sealing surfaces which, according to the single-seal tests, was the best compromise between torque and leakage values. The 0.010 and 0.020-in. deflections simulated inaccuracies in the construction of the cartridge. These deflections of 0.010, 0.015, and 0.020 correspond to seal loads of 3.9, 4.7, and 5.5 lb respectively, including the 2.12-lb seal load caused by the overbalanced area of 0.141 sq in. at a pressure of 15 psi. Figures 16, 19, 20, and 21 show the results of tests conducted at 15 psi with the four possible combinations of bellows assemblies and seal plates. Figures 17 and 18 represent the performance of bellows assembly "A" in one position at air pressures of 10 and 5 psi. The torque and leakage values obtained during the double-seal tests run at 15 psi ranged from 8.4 to 13.8 oz-in. and 15 to 65 cc per min at seal loads of 3.9 to 5.5 lb. This loading, which included the load due to the air pressure acting on the overbalanced area, corresponds to bellows deflections of 0.010 to 0.020 in. in Figures 16 through 21. These torque and leakage values are a significant improvement over the "guide" values of 34 oz-in. and 500 cc per min which are typical of seals in use at the beginning of this Phase V effort.

The torque values obtained during the double-seal tests were, as expected, approximately double the values obtained from the same seal configuration during the single-seal experiments. The leakage values obtained during the double-seal tests, however, were only slightly greater than the leakage values obtained for the equivalent single seal at corresponding loads. This unexpected lower leakage was investigated by coating the lower Teflon seal and lower seal plate (see Figure 10) with vacuum grease, and operating the test fixture as a single-seal fixture to measure only the leakage due to the upper seal. The results obtained showed that the seals making up the double-seal assembly were better than the single seals tested previously, thus accounting for the lower leakage obtained during the double-seal tests.

The repeatability of the data is evidenced by considering the data spread for the various tests run at each bellows deflection in Figures 16 through 21; as can be seen, the torque data spread is less than 2 oz-in. and the leakage data spread is less than 25 cc per min in all cases.

Considerable tolerance is available for variations occurring from construction of the seal cartridge as can be seen by comparing the torque and leakage data for the three deflections of 0.010, 0.015, and 0.020 in. For example, in Figure 16 for the 0.010 to 0.020 range of bellows deflection the leakage varied by about 30 cc per min with a maximum leakage of 50 cc per min. The torque varied by 3 oz-in. with a maximum torque of 13 oz-in. This will allow more than ample construction tolerance.

The repeatability or the ability to predict the torque and leakage values for a particular seal configuration can be illustrated by comparing Figures 16, 19, 20, and 21. This comparison gives the over-all range of 8.4 to 13.8 oz-in. and 15 to 65 cc per min, mentioned before.

The stability of the double seal is illustrated by Figures 16, 17 and 18 for bellows assembly "A" in one position at pressures of 15, 10, and 5 psi, respectively. As can

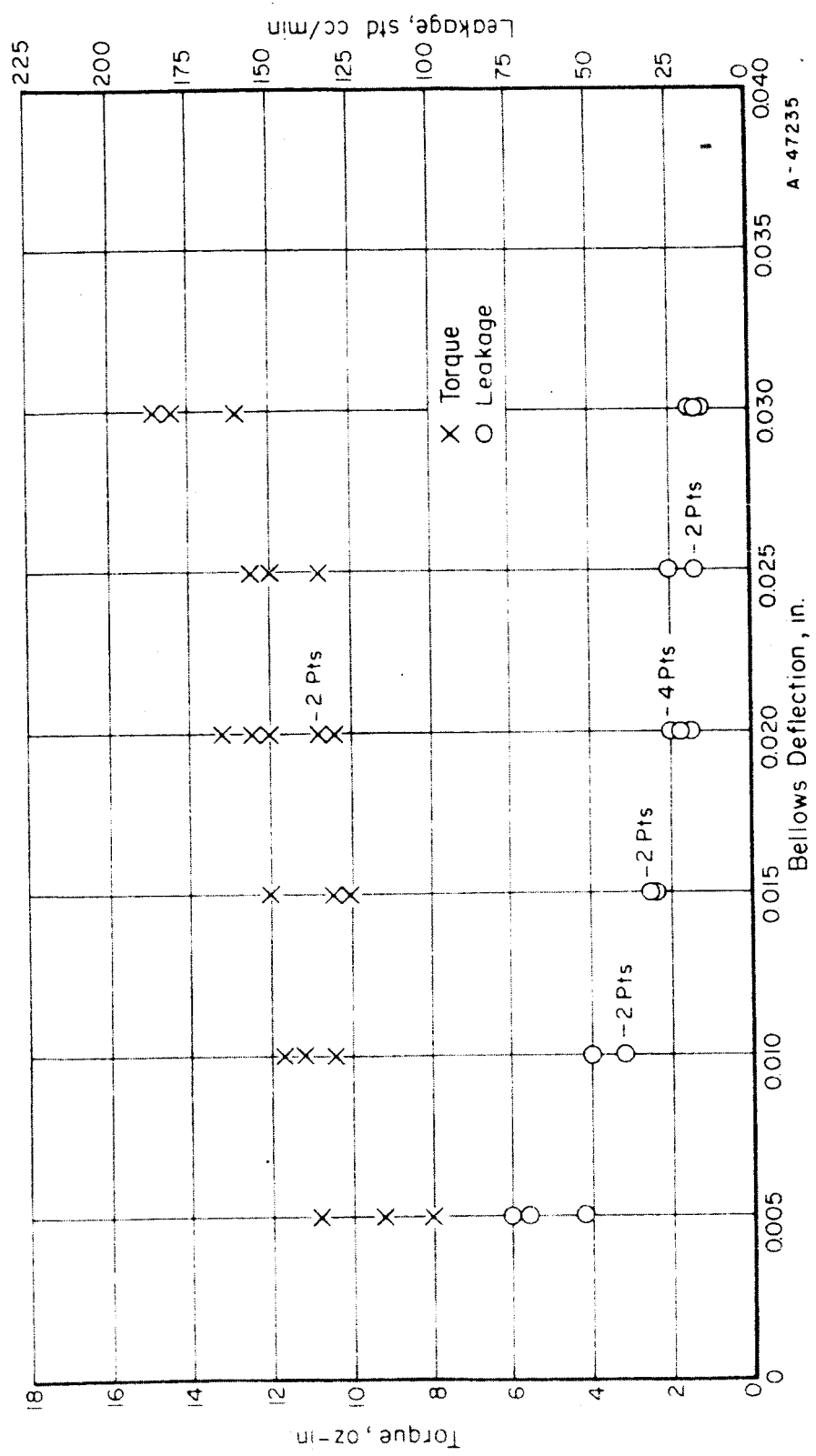


FIGURE 14. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 28

Top seal plate - 3
 Bottom seal plate - 1
 Bellows Assembly - A
 Position - right side up
 Air pressure - 15 psi.

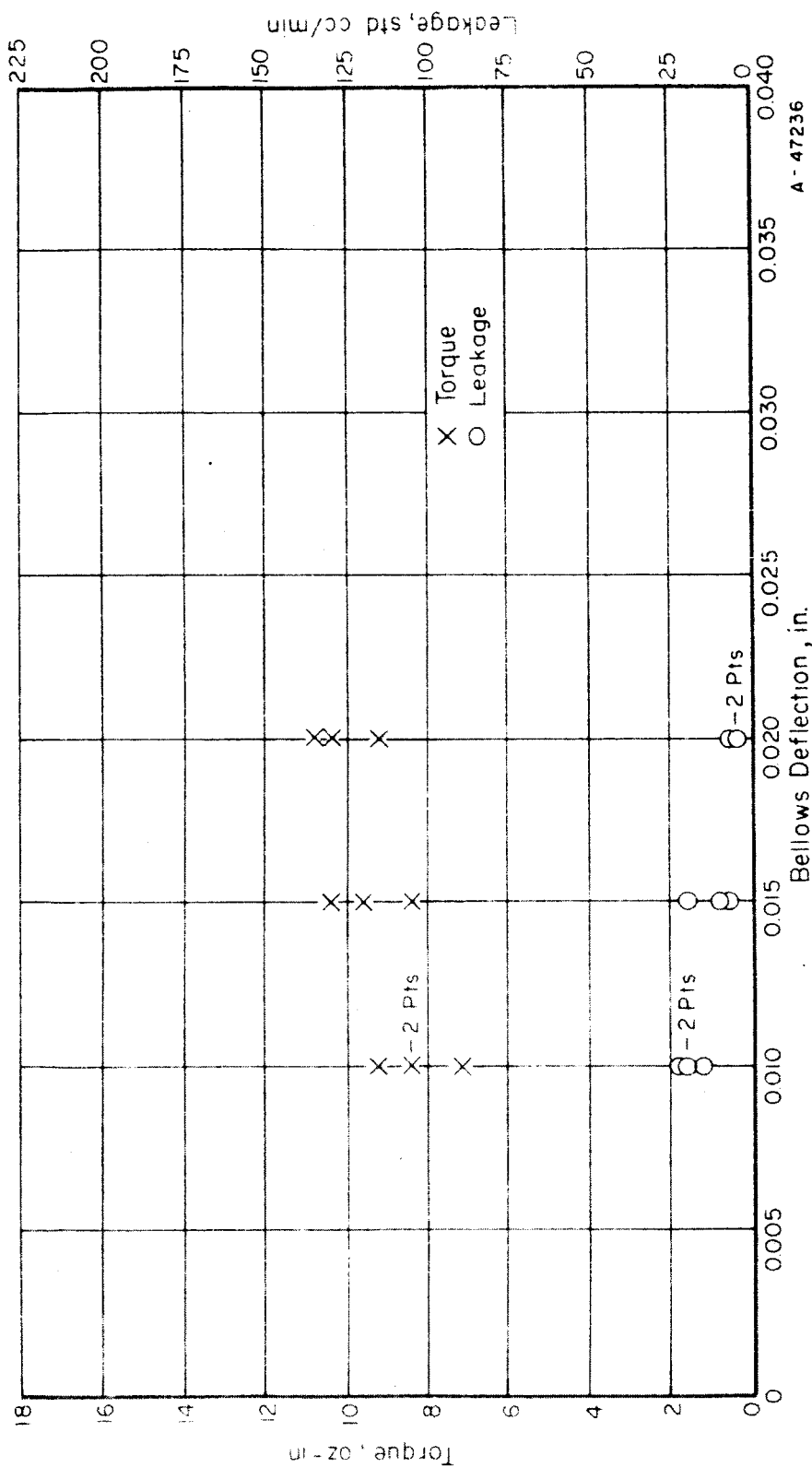


FIGURE 17. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 29

Top seal plate - 3
 Bottom seal plate - 1
 Bellows assembly - A
 Position - right side up
 Air pressure - 10 psi.

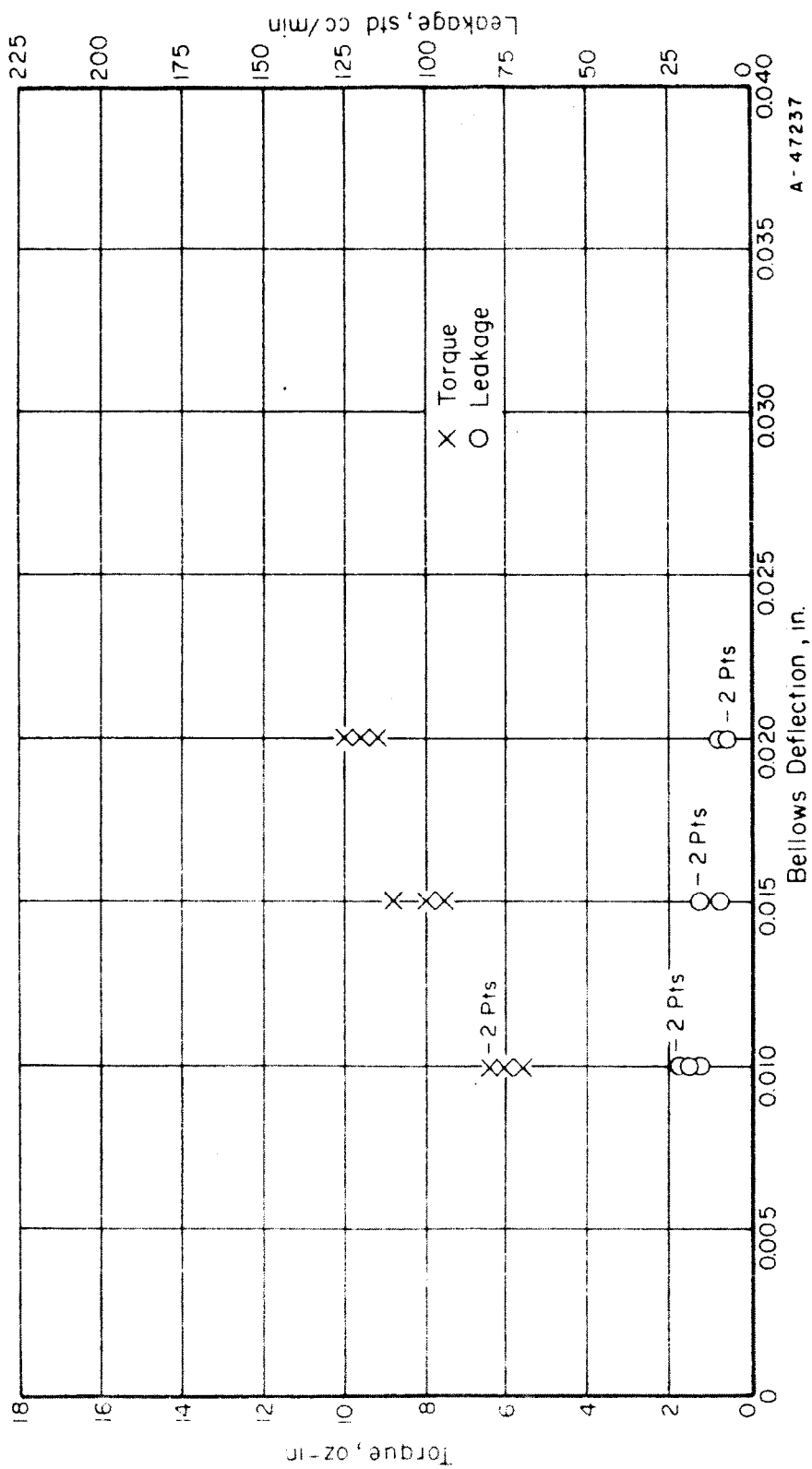


FIGURE 18. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 30

Top seal plate - 3
 Bottom seal plate - 1
 Bellows assembly - A
 Position - right side up
 Air pressure - 5 psi.

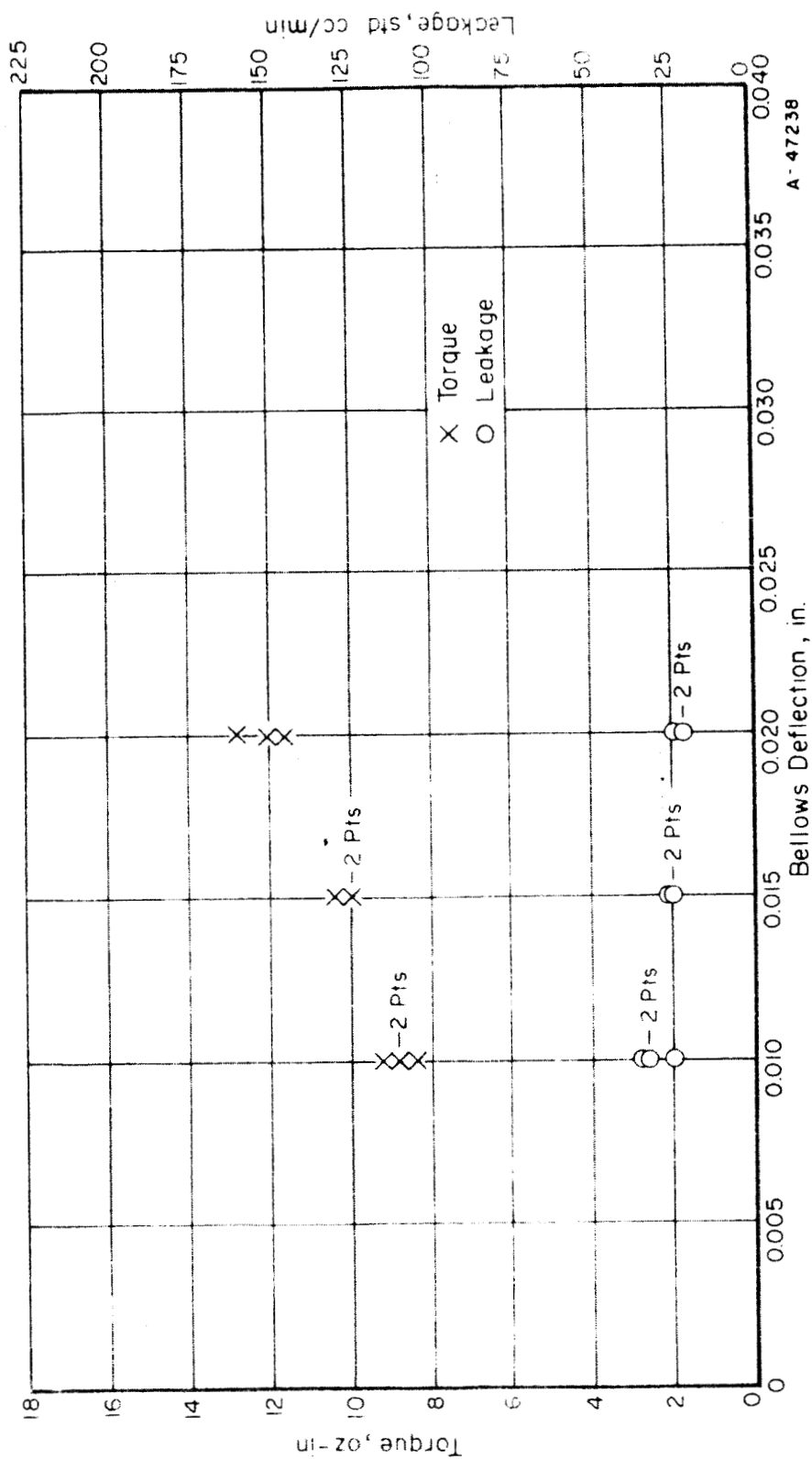


FIGURE 19. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 31

Top seal plate - 3
 Bottom seal plate - 1
 Bellows assembly - A
 Position - upside down
 Air pressure - 15 psi.

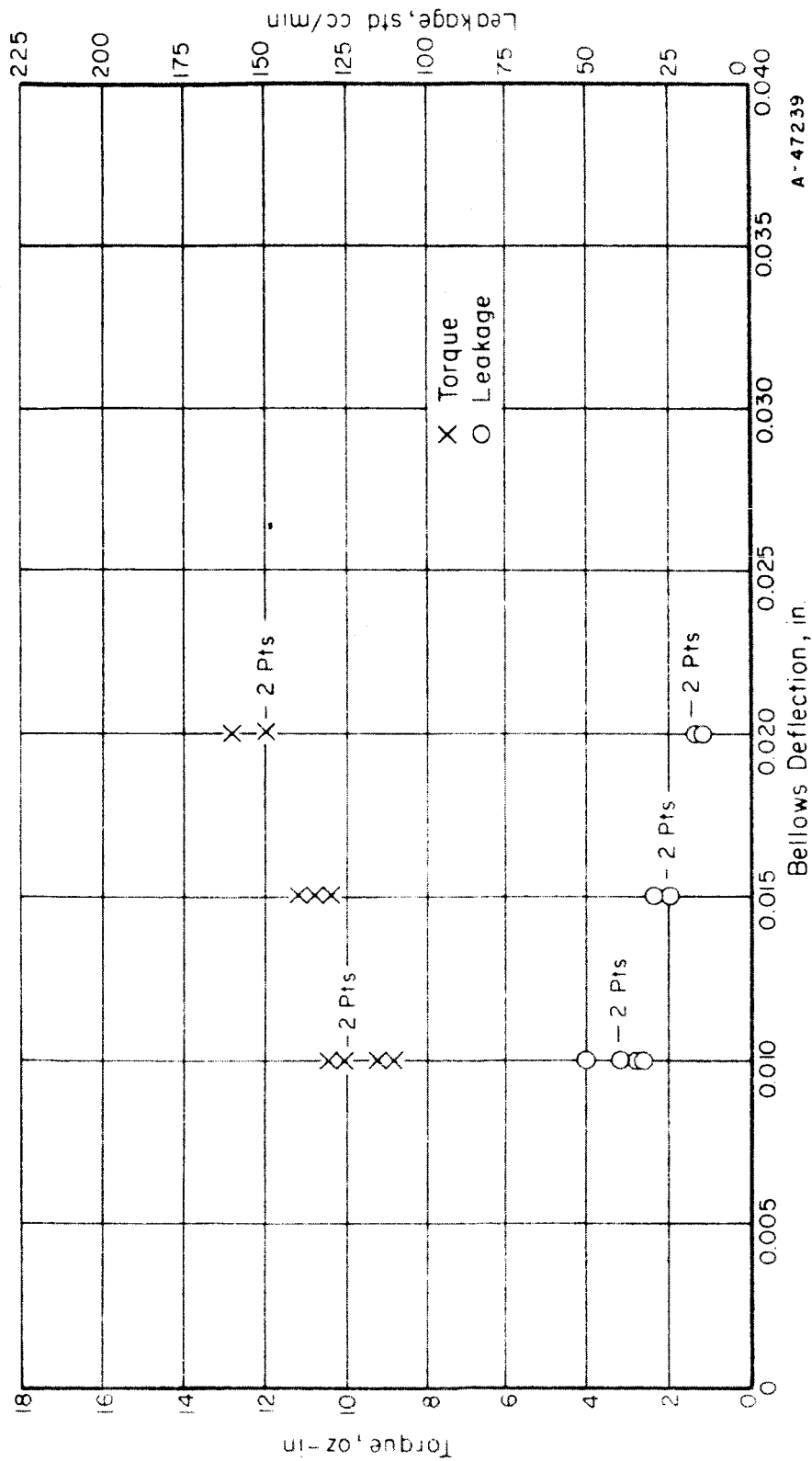


FIGURE 20. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 32

Top seal plate - 3
 Bottom seal plate - 1
 Bellows assembly - B
 Position - right side up
 Air pressure - 15 psi.

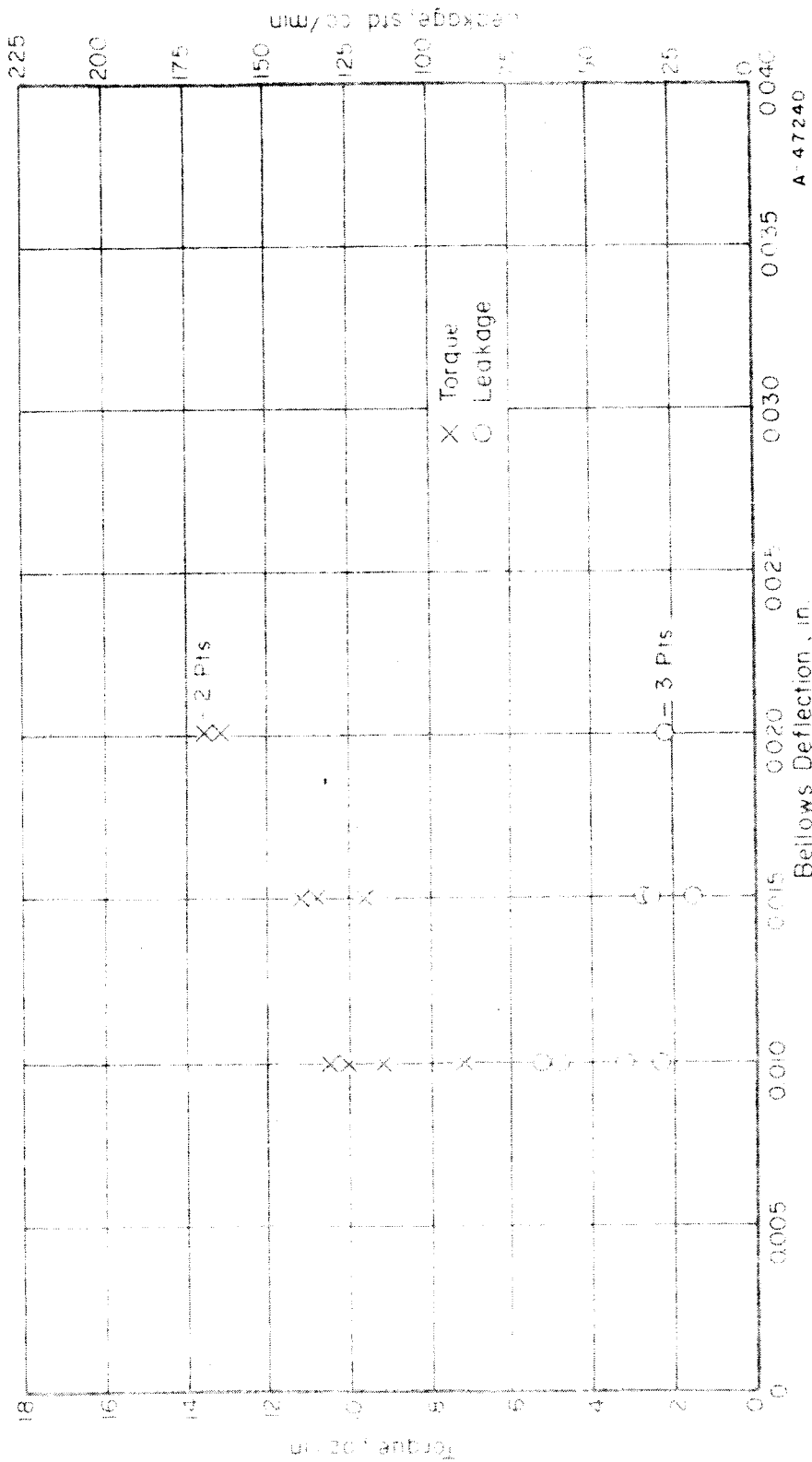


FIGURE 21. TORQUE AND LEAKAGE VS. BELLOWS DEFLECTION FOR DOUBLE SEAL-RUN NO. 3

Top seal plate - 3
 Bottom seal plate - 1
 Bellows assembly - B
 Position - upside down
 Air pressure - 15 psi.

As seen, a seal must be made completely airtight bearing in pressure. Thus, if a gimbal-seal cartridge is made of an elastic material, the seal will be leaking at the maximum pressure differential. A rigid seal, however, will be more than adequate if the pressure differential is not too great.

SEAL CARTRIDGE CHARACTERISTICS

There are two basic approaches to the design of a gimbal air seal. One approach is to assemble the seal as an integral component of the gimbal pivot during construction of the pivot. The second approach is to assemble the seal as a cartridge unit which would then be inserted in the gimbal pivot. The second approach, the design of a seal cartridge, offers considerable advantage over the component approach. The cartridge approach is discussed in the following sections.

Operating Advantages

As described in a previous section, when a Teflon seal and a mating plate are assembled, it appears to be necessary to "dress" the seal. This means that the unit must be operated for a period, disassembled, cleaned, reassembled, etc., for about three times until the dress job has been completed. If the seal components are assembled at the same time as the gimbal pivot is assembled, this assembly and disassembly would involve the entire unit. The advantage of a seal cartridge, on the other hand, would involve only one seal part. Also the possibility of the Teflon flakes (which occur during dressing) getting into the stream and being carried into the air bearings would be eliminated, with the dressing operation being done external to the gimbal system.

At the same time as the cartridge is being assembled the seal could be pretested for both radial and tangential torque characteristics. Thus it would be predetermined that the seal is ready for the test before the torque is in the gimbal system. Although it appears that the seal could be tested without critical dimensions, there is the possibility that the surface of the seal or the mating ring could be scratched during assembly or that the seal could be bent or improperly deflected. Any such errors would be detected by the test of the cartridge and might not be revealed with the seal installed in the gimbal pivot.

If and when the use of the seal cartridge is a closer specification would be possible with the seal and the mating plate. The seal would have to be built into the bearing, and the mating plate might be made of a different material.

Design Advantages

A typical seal cartridge is shown in Figure 1. The cartridge consists of the seal, the mating plate, and the gimbal pivot. The seal is attached to the torque arms and to the support of the gimbal pivot. The seal is made of a rigid material, and the dynamic seals, four static seals, are used.

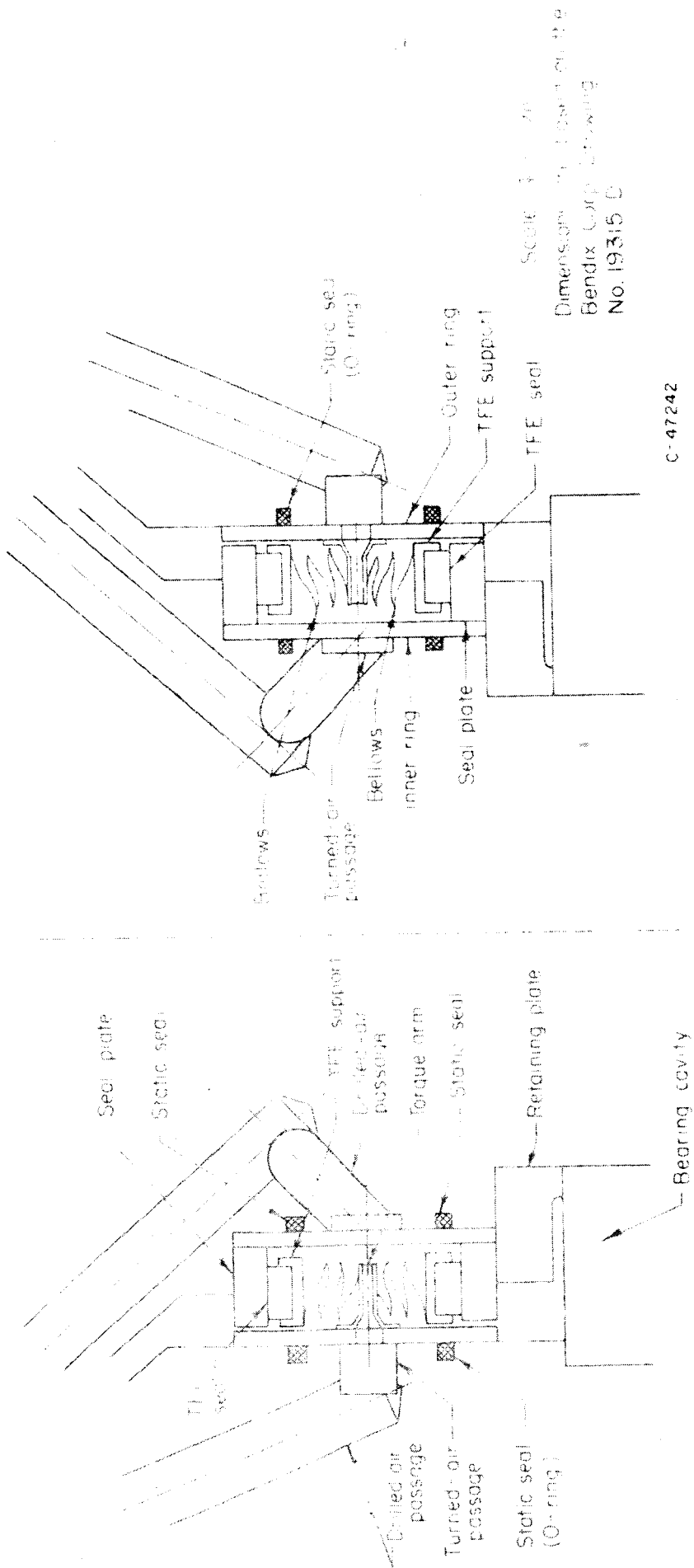


FIGURE 22. TYPICAL CARTRIDGE SEAL.

One of the most important benefits of this principle is the fact that the bellows member of the seal can be designed so as to distribute the force on the sealing surfaces. This can be accomplished by making the bellows arm flexible in the axial direction, but resistant in the circumferential direction. This is a help in widening the construction tolerances. The floating principle also allows the spring rate of the bellows to be sufficiently high to achieve a high natural frequency of the bellows/mass system, thus preventing resonance.

The mean diameter of the Teflon seal should, of course, be made as small as possible in order to reduce the torque. In the Linnal system as defined by The Bendix Corporation Drawing No. 19315D, which is a layout of the plus Z pivot, the average seal diameter is closely limited. The possible cartridge shown in Figure 22 is based on the construction and dimensions shown in The Bendix Corporation Drawing No. 19315D.

In Figure 22 the inner ring and seal plates are pressed together on assembly to obtain the proper deflection of the bellows and create an enclosed cartridge, thus eliminating the possibility of a scratch or gouge on the sealing faces. The inner and outer rings are pressed into the pivot together after the four static seals have been placed in the pivot compartments.

The major reason for using a bellows as a spring member is twofold. (1) the welded metal bellows provides a constant pressure wall, and (2) it provides an even load application on the Teflon sealing member. Teflon supports are provided so that an even distribution of load is transferred to the Teflon seal. An uneven loading on a Teflon seal may cause a warpage of the Teflon which would probably result in excessive leakage.

The inner and outer seals do not contact rings. It might be possible to size these to allow sufficient axial movement of the outer ring that a floating condition would be provided in the bellows and torque arm assembly. However, a more reliable floating condition can be achieved only by specially designed torque arms as described above.

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